POWER

Volume 60 New York, September 2, 1924 *No.* 10

Learning from Life

YOU would like to try it over again—to have another chance, wouldn't you?

This is true of so many things, from a stroke at golf to the great game of life itself.

You have played it once with more or less success, but, however well you have done, you know now just how it should have been played and if you had but another chance——

For games and deals and ventures of one kind and another more chances will come, and opportunities will offer to profit by observation and experience.

But one can live his life but once, and everybody is convinced, as the score begins to round up and the net result to show, that he could have made so much more of it if he had only known.

I recently attended an impressive function in honor of a great man and took the first opportunity to read his biography.

And the thought came to me as I read, that here was an opportunity for getting, in advance and while opportunity still offered for its application, some of the hindsight, some appreciation of the elements of success, some of the habits of thought and accomplishment that make men useful and efficient and enable them to do things that make the world better for their having lived in it.

One cannot re-live his own life but he can review the struggles, methods and accomplishments of those who have succeeded, and, warmed and inspired by their experience, perhaps make better use of the time and opportunities that are still ahead of him.

F. R. Low

Water-Cooled Furnaces Make Records at Hell Gate

Side Walls Made of 4-In. Tubes with Overlapping Fins—Furnace Deterioration Greatly Reduced—Combined Efficiency of 92.7 per Cent Obtained

By W. E. CALDWELL

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IN THE settings of the twelve boilers comprising the initial installation at Hell Gate Station, a double-ended furnace is employed with a short stoker firing at each end. At the time of preparing this layout the short stoker had reached a high state of development, while the long stoker was still in the experimental stage, which consideration prompted the design adopted. The stokers have a length of 17 tuyeres and a width of

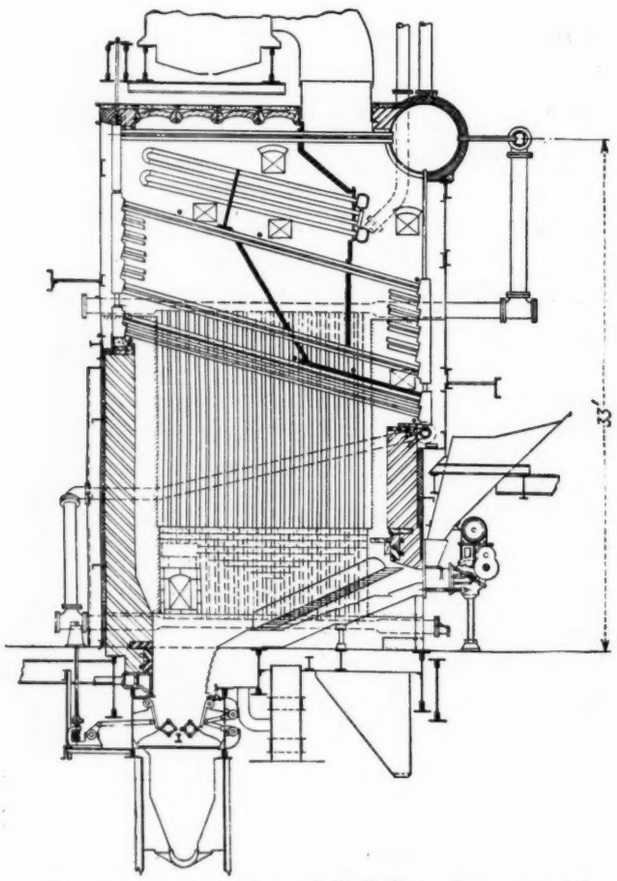

Fig. 1—Cross-section of Hell Gate furnace with water-cooled side walls

14 retorts at each end of the boiler, making a total of 28 retorts per boiler. The boilers are 20 tubes high with the superheater in an interdeck location, that is, with six rows of tubes between the superheater and the furnace. No economizers were installed in the original installation, although room was provided for future insertion should conditions warrant it. Each boiler is equipped with an induced-draft fan driven by an alternating-current motor of the brush shifting type.

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The foregoing general description is included for comparison with the more recent installation. For a more detailed description of the original boilers and their settings, consult the May 9, 1922, issue of *Power*.

When load conditions necessitated an increase in station capacity, turbo-generator unit No. 5 was installed, together with three additional boilers known as No. 51, No. 52 and No. 53. These three new boilers, while of the same make as the original installation, differed somewhat in design, especially as regards furnace design, baffling and superheater location. Perhaps the most pronounced departure from conventional practice was the installation of side-wall cooling tubes in the furnaces,[1] the general arrangement of which is indicated in Figs. 1 and 6.

These side walls consist of 4-in. tubes arranged vertically on each side of the furnace and spaced on 7-in. centers. Each tube has two longitudinal steel fins welded on it diametrically opposite each other, and when placed in the boiler furnace, the fins overlap, thus presenting a continuous water-cooled surface to the radiant heat of the furnace. The lower portion of the tube is covered by fireclay tile for a short distance above the stoker. This extends along a horizontal line the entire length of the furnace.

The water in the tubes forms a part of the regular boiler circulation, the lower end of the tubes being connected by headers leading to the mud drum. The upper end of the side-wall tubes are connected to a header, which in turn discharges into the boiler drum through a special set of nipples and headers in front of the drum.

The water-cooled side walls not only replace practically all the brickwork in the side walls, but afford considerable protection to the front and rear walls by absorbing the heat which they reflect and thus lowering the temperature of the face of the wall.

To anyone familiar with the difficulties of brickwork maintenance, the logic of the water-cooled walls is at once evident. Not only is the problem of brickwork maintenance in large boiler settings an item of considerable expense, but the loss of capacity of steam-generating equipment rendered idle by necessary repair periods is of even greater importance. In the older plants in which smaller boilers and furnaces are employed, it is not unusual to complete a major brickwork repair job over a week-end or holiday period. With the larger settings several days are required, and each boiler represents a larger percentage of the total steaming capacity than it did in the older stations having a larger number of small boilers.

The furnaces in these new boilers are of the single-ended type as contrasted with the double-fired furnaces of the original installation. The long stoker had passed

[1]These furnaces were designed by Thomas E. Murray, consulting engineer, New York City, the original designer of Hell Gate Station. The patents are now held by Combustion Engineering Corporation.

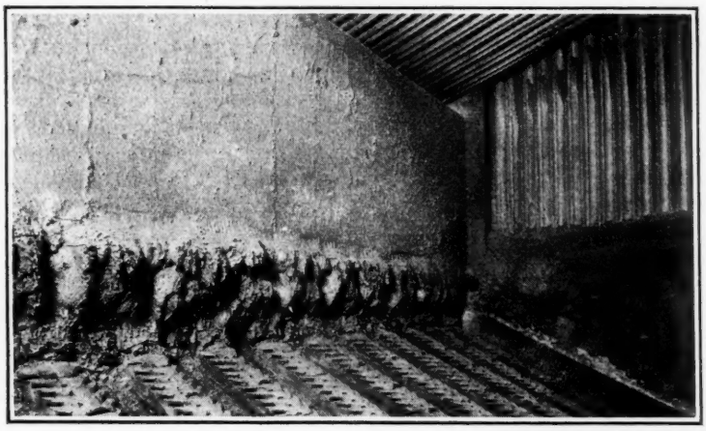

Fig. 2—Interior of furnace after several months' operation, showing front wall and fin tubes of side wall.

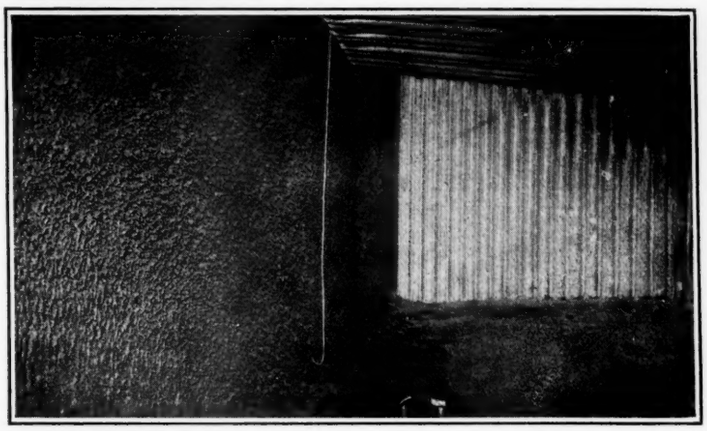

Fig. 3—Interior of furnace, showing rear wall and side wall after several months' operation.

Fig. 4—Gas sampling and analyzing apparatus set up for testing boiler.

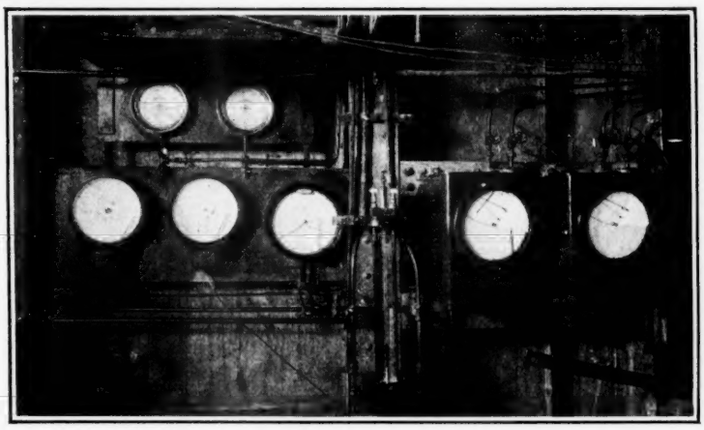

Fig. 5—Instruments recording draft, air flow, steam flow, pressures and temperatures.

through the experimental stage since the installation of the first stokers at Hell Gate, and while it was not in the finished state of its predecessor it possessed certain advantages that seemed to outweigh any objections on this account. Stokers of the same make were installed, having a length of 33 tuyeres and a width of 14 retorts and employing double-roll clinker grinders as in the original installation, but with certain minor changes. Many other changes, found advisable from past experience, were made in the air piping and arrangement. The superheaters are of the straight convection type and are located between the first and second pass, as shown in Fig. 1. These boilers are provided with economizers. The induced-draft equipment is the exact duplicate of that employed on the older installation. The new boilers have now been in

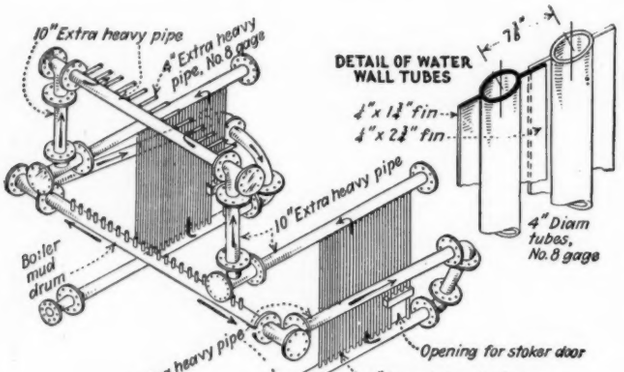

Fig. 6—Isometric sketches showing detail of fin tubes and method of connecting into boiler circulation

service several months, and from both a performance and maintenance viewpoint the results have been very gratifying.

Fig. 2 is a photograph showing the interior of the furnace after several months' operation. The water-cooled tubes and fins had not been cleaned at the time this photograph was taken. It may be noted that the front wall also appears to be in an excellent condition except for a very small zone adjacent to the ram boxes.

Fig. 3 shows the interior of the furnace looking toward the rear wall and side walls. The clean appearance of the side-wall tubes and rear wall contrasts sharply with the clinker accumulations which are usually seen adhering to the brick walls in the older settings.

From indications, it is evident that this new type of setting will increase the life of the front and rear walls two or three times. The only evidence of deterioration has been on the tile covering the lower part of the side-wall tubes. In future designs the elimination of this tile is contemplated.

Condensed Summary of Tests on No. 51 Boiler

Soon after the first of these three new boilers was placed in service, tests were conducted to determine its efficiency and capacity. It so happened that Boiler No. 51 was the first to be placed in operation, and as the three boilers are identical in every respect, the performance of this boiler is representative for the particular design adopted.

The following remarks and the accompanying test data were presented by the writer before the Metropolitan Section of the A.S.M.E. at the April meeting in New York City.

The tests on this boiler unit were conducted by J. J. Grob, Test Engineer of the United Electric Light & Power Co., and were witnessed by John Blizard of the Power Specialty Co. We are also indebted to Mr. Blizard and the company he represents for the measurements of gas temperatures which were taken under

TABLE I—CONDENSED SUMMARY OF TESTS ON NO. 51 BOILER

Boiler-heating surface including area of side wall, tubes and fins exposed to furnace, sq.ft.			15,900
Superheater surface, sq.ft.			7,430
Economizer surface, sq.ft.			13,824
Grate surface, sq.ft.			378
Furnace volume, cu.ft.			8,000
Test Number	3	5	Max. Rat.
Date 1924	Feb. 25-26	Mar. 18-19	Feb. 15
Duration, hours	25	28	¾
Coal fired per hour (as received), lb.	7,633	8,133	.....
Actual water per hour, lb.	87,750	88,221	265,000
Equivalent water per hour, boiler and superheater, lb.	93,278	99,355	289,000
Equivalent water per hour, boiler and superheater, and econ., lb.	103,809	106,003	308,275
Quality of saturated steam, per cent	98.75	98.89	98.10
Pressure in boiler drum, lb. gage	267	266	280
Pressure in superheater outlet, lb. gage	263	263	244
Temperatures (Degrees F.)			
Feed water entering economizer	205	128.2	165
Feed water entering boiler	285	233.1	267
Boiler room	65	81	.....
Air for combustion	58	65	.....
Flue gases leaving boiler	455	485	660
Flue gases leaving economizer	229	193	336
Superheated steam	547	565.3	563
Draft (inches of water)			..
Drop through boiler and superheater	0.45	0.50	3.27
Drop through economizer	0.21	0.27	2.00
Total draft loss	0.66	0.77	5.27
Forced-draft pressure in windbox	1.16	1.85	6.00
Rating developed by boiler and superheater, per cent*	170	181	528
Combined efficiency, per cent — Boiler and superheater	83.2	84.6	.....
Combined efficiency, per cent — Boiler, superheater and economizer	89.6	92.7	.....
Proximate Analysis			
Test number		3	5
Moisture, per cent		2.55	3.80
Volatile matter, per cent		20.00	18.85
Fixed carbon, per cent		70.42	70.19
Ash, per cent		7.03	7.16
Fusion temp. deg. F.		2,440	2,450
B.t.u. per lb. as fired		14,261	14,010
B.t.u. per lb. dry		14,626	14,563
B.t.u. per lb. combustible		15,760	15,780
Ultimate Analysis			
Sulphur, per cent		0.67	0.77
Hydrogen, per cent		4.49	4.32
Oxygen, per cent		4.19	4.12
Nitrogen, per cent		1.06	1.07
Carbon, per cent		82.35	82.28
Ash, per cent		7.22	7.44
Ash			
Refuse, per cent of coal fired		7.46	7.80
Per cent combustible		4.50	4.44

Flue Gas

Test number	3		5		Max. Rating
	Leaving Boiler	Leaving Econom.	Leaving Boiler	Leaving Econom.	Leaving Boiler
Carbon dioxide, per cent	11.69	10.91	13.04	12.01	15.2
Oxygen	7.52	8.45	6.25	7.28	0.8
Carbon monoxide, per cent	0.06	0.04	0.04	0.03	3.3
Pounds of gas per lb. of coal	17.55	18.78	15.83	17.13	....
Per cent of excess air	60.1	71.6	38.4	49.7	....

*Since these data were presented before the Metropolitan Section of the American Society of Mechanical Engineers, an eleven-hour test was made to determine the maximum capacity that could be carried for this period. The average rating obtained was 461 per cent, with peaks up to 603 per cent. The limiting factor was the inability to feed more coal and remove the gases with the existing equipment.

his direction. These temperatures were taken at four points across the boiler outlet and four points across the economizer outlet in order to obtain an average for each location. Thermocouples of the type recommended by the Bureau of Mines were used for this purpose in connection with a potentiometer. Immediately after the tests the couples were tested by the Bureau of Standards and their indications were found to be accurate to within plus or minus three degrees.

The tests here reported were selected from several that were conducted shortly after the boiler was placed in service. The usual care was exercised to secure high accuracy. The coal was weighed by weighing lorries, the accuracy of which was previously checked

with dead weights. The water evaporated was measured by means of a venturi meter and mercury manometer standardized before and after the tests by the use of weighing tanks. The final check on the venturi meter was substantially the same as the initial check, and in no case was the difference between the weighed water and the flow shown by the manometer, when corrected for temperature, greater than 0.3 per cent. The steam flow from the boiler was also measured by a Bailey meter and used as a check against the measured water.

In conducting tests No. 3 and No. 5, the feed-water flow was maintained constant by regulating the feed valve in accordance with the indications of the manometer. The level of the water in the boiler was in turn controlled by regulating the combustion air supplied by the forced-draft fans in accordance with the height of the water in the gage glass. This was accomplished simply by maintaining the air-duct pressure slightly in excess of that required by the average rating, and slight fluctuations were taken care of by means of the hand damper. In other words, the flow of water to the boiler was fixed, while the amount of heat imparted to the boiler was regulated to maintain an average rate of evaporation corresponding with the rate at which the feed water was admitted. The steam pressure was controlled by means of the other boilers in the station.

In the maximum rating test the capacity of the venturi meter was exceeded, and the rating was obtained by means of a manometer-type steam-flow meter.

Fig. 4 shows the gas-sampling pipes and some of the apparatus used in analyzing flue gas during the tests. Gas samples were taken from three points in the boiler outlet and three points in the economizer outlet, by means of a steam-jet aspirator and bubbling bottles with which the flow was regulated. A Mono recorder also took a continuous sample from the boiler outlet.

TABLE II—HEAT BALANCE OF TESTS 3 AND 5

Boiler and Superheater

Test Number	3		5	
	B.t.u.	Per Cent	B.t.u.	Per Cent
Heat per pound of dry coal	14,626	100	14,563	100
Absorbed by water and steam in boiler and superheater	12,153.1	83.2	12,322.1	84.6
Loss due to moisture in coal	32.3	0.2	48.2	0.3
Loss due to water from combustion of hydrogen	512.4	3.5	492.6	3.4
Loss due to moisture in air	7.9	0.1	1.2	+0.0
Loss due to dry chimney gases	1,642.7	11.2	1,420.9	9.8
Loss due to incomplete combustion of carbon	41.7	0.3	25.4	0.2
Loss due to unconsumed combustible in refuse	62.8	0.4	52.6	0.4
Loss due to unconsumed hydrogen and hydro-carbons, radiation, and unaccounted for	173.1	1.2	200.0	1.4
		100.1		100.0

Boiler, Superheater and Economizer

Temperature of feed water entering economizer, deg. F	205.0		128.2	
	B.t.u.	Per Cent	B.t.u.	Per Cent
Heat per pound of dry coal	14,626	100	14,563	100
Absorbed by water and steam in boiler and superheater	12,153.1	83.2	12,322.1	84.6
Absorbed by water in economizer	956.0	6.5	1,183.2	8.1
Loss due to moisture in coal	29.7	0.2	43.7	0.3
Loss due to water from combustion of hydrogen	470.9	3.2	446.7	3.1
Loss due to moisture in air	3.4	+0.0	2.5	+0.0
Loss due to dry chimney gases	739.2	5.1	460.4	3.2
Loss due to incomplete combustion of carbon	30.4	0.2	20.7	0.1
Loss due to unconsumed combustible in refuse	62.8	0.4	52.6	0.4
Loss due to unconsumed hydrogen and hydro-carbons, radiation and unaccounted for	181.0	1.2	31.1	0.2
		100.0		100.0

Fig. 5 shows the Bailey meters together with indicating and recording pressure gages and thermometers. The recording thermometer gives a record of the temperature of the air for combustion. The differential pressure recorder is connected in with the venturi manometer and serves as a check on the operator in charge of the feed valve during the test. The recording pressure gage at the extreme left records the pressure in the boiler drum, while the two indicating gages above indicate the steam pressure at each superheater outlet.

In comparing tests Nos. 3 and 5, it may be seen that, aside from the slight difference in rating, the difference in over-all efficiency is due to lower feed-water temperature to the economizer and a reduction in the amount of excess air in the latter test. The steam temperatures were somewhat lower than desired due to the introduction of the water-cooled surface in the side walls and the consequent lowering of the gas temperature at the superheater.

The difference between the composition of the gas leaving the boiler and that leaving the economizer was due largely to leakage in the expansion joint between boiler and economizer and other points that had not been properly finished at the time of conducting the tests. The tests were conducted shortly after the boiler was placed in operation when it was not in a finished condition. Subsequent to these tests air infiltration has been practically eliminated.

In referring to the flue-gas analysis recorded during the maximum-rating test, it may be noted that a large quantity of carbon monoxide was present. At such high coal-burning rates, with the type of equipment, it is practically impossible to avoid the loss due to the escape of carbon monoxide unless air is admitted above the fuel bed. In this particular installation, however, it is rarely necessary to carry such high rating, hence the efficiency at this capacity is relatively unimportant.

During the conduct of the tests no furnace temperatures were taken, but from rough observations it appeared that the temperature of the fuel bed was no lower than results from the corresponding rating on boilers not equipped with water-cooled side walls.

At some distance above the fuel bed the flame appeared more transparent than in boilers not equipped with the side-wall tubes, which seemed to be due to the absence of red-hot reflecting surfaces on the opposite side from the point of observation. These observations apply to the tests at lower rating, but inasmuch as such favorable performance was shown, any effect of the water-cooled tubes on the temperature of the furnace or fuel bed could hardly have had any ill effect on the combustion conditions.

At the higher rating the temperature of the fuel bed and furnace appeared to be about the same as at equivalent rating with other boilers not equipped with side-wall tubes, although that portion of the brickwork and side walls which was visible from the observation door had a decidedly lower temperature than exists in the older equipment at much lower rating.

While observations may not be entirely trustworthy under such conditions, it is felt that they are partly borne out by the performance of the unit and by the known reduction in brickwork maintenance, based on past experience.

If we correct the efficiency results of Tests No. 3 and No. 5 for the power consumption of the stoker, fans and other power-consuming apparatus in connection with the boiler, there will of course be a slight reduction

in the over-all efficiency. For example, the induced-draft fan required about 25 hp. electrical input to the driving motor, and as the forced-draft fan motor required a similar amount, we have a total of approximately 50 hp. required by the fans. The stoker and clinker grinder required another 10 hp., making a total of 60 hp. as an outside figure for the auxiliaries of the boiler proper. We may ignore the power consumption of the boiler feed pump, since this is an item common to all plants, and one that would not be influenced greatly by the design of the boiler. Inasmuch as the station coal rate is less than one pound of coal per horsepower-hour, the coal equivalent of the energy consumed by the boiler auxiliary apparatus is roughly 60 lb. of coal per hour, or approximately 0.8 per cent of the coal burned during these tests. Hence the net combined efficiencies may be obtained by deducting 0.8 per cent from those reported in the table, which still leaves an over-all net efficiency in Test No. 5 above 90 per cent.

Now to compare these figures with other tests which are sometimes reported on the low calorific value of the coal, or corrected for the resulting loss, which accompanies the formation of water vapor by the burning of hydrogen, we may simply add the amount of the hydrogen loss as shown in the heat balance, or approximately 3.2 per cent, to the figures reported in the table.

It is believed that the results here presented establish a new record of performance for this type of equipment, not only on the over-all efficiency basis, but also from the standpoint of total steam produced by a single unit.

New Type of Butterfly Valves Used on Davis Bridge Hydro-Electric Project

An improved type of butterfly valve, known as the Dow disk-arm pivot valve, has been extensively used on the New England Power Co.'s Davis Bridge project and was developed by E. A. Dow, mechanical engineer for that company. It represents an attempt to retain the simplicity and moderate cost of the usual type of butterfly valve and at the same time provide an operating mechanism of such type and rigidness as to increase reliability and reduce leakage. These results are obtained by the use of an arm or lug cast integral with the disk, by means of which a connecting rod can apply the operating force directly, rather than through a crank keyed to the spindle, as has been customary. This affords an operating mechanism independent of the torsional stiffness of the spindle and of such a nature that it can be built of any desired strength.

The new valve was first developed for the Davis Bridge reservoir intake, the general design of which was described in the May 13 issue of *Power*. The arrangement of one of the two valves in the intake structure to the 14-ft. tunnel is shown in Fig. 1, and a photograph of one of the valves, which are 96 in. in diameter, is shown in Fig. 2.

The reservoir is used for seasonal storage and the unusual fluctation of over 90 ft. in the water level necessitated the use of some type of gate or valve that could be operated satisfactorily under heads varying from 16 to 110 ft. The importance of the installation made reliability of first importance, and in order to keep the cost of the tower leading to the valves within reason a compact design was required. Accessibility of operating mechanism and the ability to make repairs or replacements with minimum interference with operation were also important considerations, and the head loss had to be kept moderate.

It is believed that the design adopted satisfactorily meets these requirements. The tower is of the dry type, all mechanism being accessible at any time. The use of two separate valves, either of which can, in an emergency pass the total required quantity of water and either of which can be removed by the use of upstream and downstream bulkheads, insures continuity of operation while any repairs are being made.

The most vulnerable feature of the valves, the seats, are in this case made of babbitt cast and peened into a groove in the housings, which are bored to exact size. This is a new type of seat and if damaged can

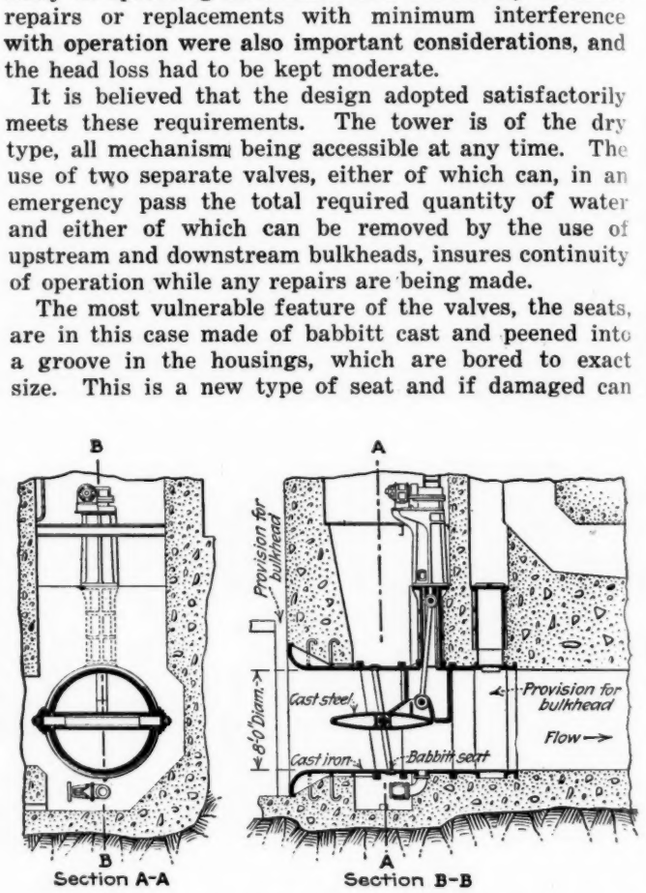

Fig. 1—Sections through disk-arm valve at entrance to power tunnel

be repaired in place with little difficulty and expense, it being merely necessary to fuse additional material onto the damaged spot with a torch and scrape it down to match the adjacent bore of the cast-iron housing. The operating mechanism being on top permitted the closest possible spacing and resulted in the minimum size of tower. An equally close spacing with the usual type of butterfly valve would have been possible only by placing the spindles vertical, which was considered undesirable because of the difficulty of supporting the disk in exactly the correct position to avoid wear of the seat as well as the difficulty of preventing sand from getting into the lower bearing with the resulting cutting.

The operating mechanism is so proportioned that the valves can be safely closed under the maximum velocity possible with a ruptured penstock or a blowout in the tunnel. When closed, they are tight, since the thrust of the hoist is so applied as to overcome the tendency of the lower half of the disk to deflect downstream and open up a crack around the lower semi-circumference. Shop tests of these valves showed a leakage of 1¾ gal. per minute at 160-ft. head. Observations in place under 110-ft. head showed one valve to be absolutely tight, while the other leaked 5 to 6 gal. per minute. No snap rings or other staunching devices are necessary.

The successful development of the disk-arm design

for the intake led to its adoption at the upper ends of the penstocks, where the static head reaches 160 ft., which the surge incident to a shutdown increases to 200 ft. Sturdy construction was particularly attractive here on account of an accident that occurred in a similar location at another one of the New England Power Co.'s plants. In that instance a thick layer of ice, which had built up on the inside of a penstock, loosened in the spring, floated to the top of the pipe line and struck a butterfly valve with such force that the operating arm on the end of the spindle failed and allowed the disk to slam closed. By the use of the disk-arm mechanism sufficient strength has been obtained to render damage from such an occurrence unlikely.

The operating mechanism of the penstock valves differs somewhat from that of the intake valves, the hoists being mounted on trunnions which permit them to incline as the valve is opened or closed. This eliminates the necessity of a crosshead and guide, and in this instance resulted in a reduction in cost. The joint where the operating rod passes through the housing is made tight by the use of a ball-and-socket packing. Fig. 3 shows one of the valves in place at the upper end of one of the penstocks. These valves also are designed to close if necessary under full spouting velocity. The hoists permit tight closures, and the measured leakage in place under 145-ft. head was about 2 gal. per minute each.

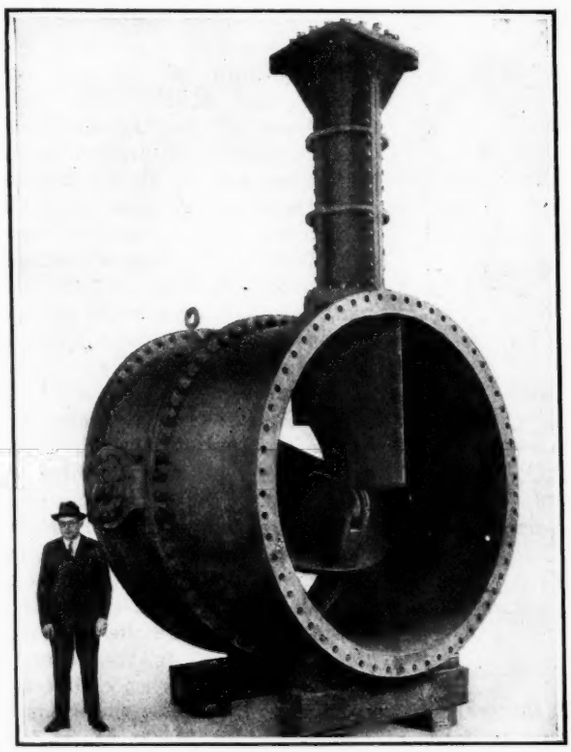

Fig. 2—Davis Bridge intake valve, 96 in. in diameter

Probably the most interesting application of the disk-arm valve at Davis Bridge is in their use for a free discharge bypass through the dam under a head of about 200 ft. For this purpose a 48-in.- and a 36-in. valve are used in series to pass water through the plug in the tunnel, which was constructed to handle the flow of the river during the construction of the dam. The placing of the plug necessitated the temporary use of the 48-in. valve, but only a 36-in. size was needed as a permanent bypass to supply the plants below in an emergency, hence the reduction in size. The small valve will furnish the primary control, the other being used for shutoff purposes if repairs are necessary to the 36-in. one. Both valves are operated by reservoir pressure. The 36-in. valve under shop test showed a leakage of one quart per minute under a 350-ft. head, and is equally tight in service.

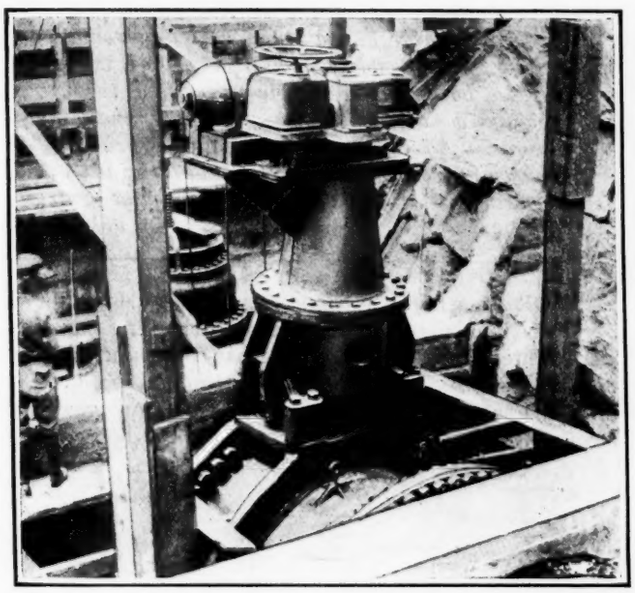

Fig. 3—Operating mechanism of penstock valves mounted on trunnions

Since these valves will have to work under conditions to which only needle valves have usually been applied, it was thought desirable before they were adopted to make tests to determine the action of butterfly valves under high velocity, no data on the subject being available. Tests were run on a 78-in. valve under 240-ft. head at the New England Power Co.'s plant No. 5 and also on an 8-in. model valve. The results showed that, instead of being balanced under all conditions, a butterfly valve has a decided closing tendency at all except wide-open and full-closed positions, if there is any flow through it. For high velocities this tendency is sufficient to endanger the control mechanism of the ordinary butterfly as commonly proportioned. Another attractive feature about the disk-arm valve for free discharge service is that when it is fully open the disk arm is drawn tightly against the projecting stem shield inside the housing, thus providing a rigid buttress, which eliminates vibration.

The disk-arm valve proved so attractive after being fully developed that its originator has applied for patents and it is now being offered for general use, being manufactured in sizes up to 6 ft. in diameter by the Coffin Valve Co., Boston, Mass., and in larger sizes by S. Morgan Smith Co., York, Pa. Two important installations in addition to those at Davis Bridge have been in service for several months. One of these consists of six 72-in. valves used as free discharge flood gates under 110-ft. head. The other is a set of six 78-in. valves bypassing water through a power-house substructure under a head of 40 feet.

Cross-Compound Impulse Turbine Geared to 4,000-Kw. D.-C. Generator

A reduction gear is the logical connection between a turbine and a large direct-current generator. The principle is here developed so as to include both high- and low-pressure turbines, driving a common gear through individual pinions. Advantages regarding material, efficiency and reliability may be thus attained over those characteristic of a single-cylinder geared turbine.

FOR exacting conditions of geared turbine service, such as marine propulsion, furnishing municipal water supply, etc., the cross-compound principle has been utilized with advantage. In driving a 4,000-kw. direct-current generator, where reduction gears are ordinarily required, this arrangement has been adopted as shown in Fig. 1, utilizing three flexible type couplings.

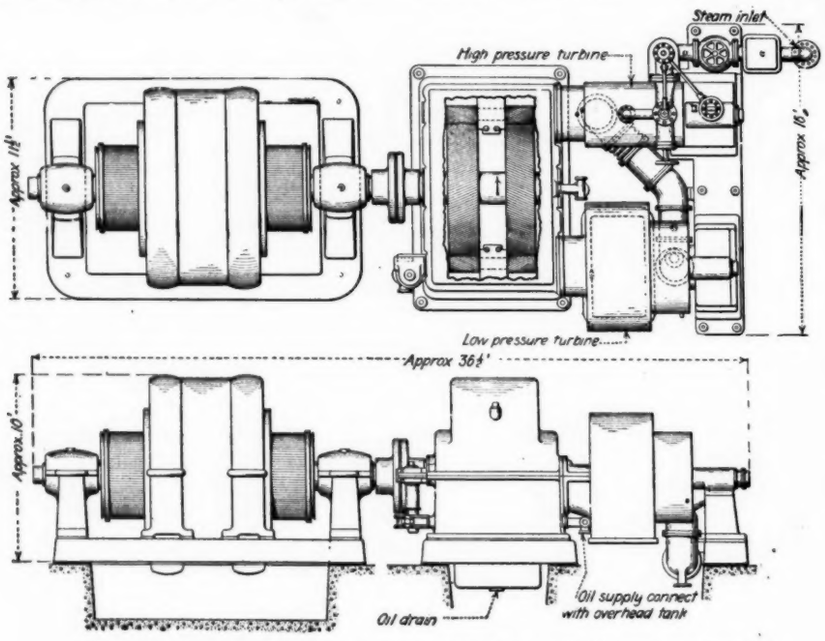

Fig. 1—High- and low-pressure cylinders supply approximately equal power at normal load

Three of these geared cross-compound direct-current De Laval units, as mentioned in the May 27, 1924, issue, of *Power* have been installed for auxiliary drive in the Trenton Channel Plant of the Detroit Edison Company. Each unit contains two direct-current generators operating in parallel at 250 volts, capable of delivering 4,000 kw. at normal load and 6,000 kw. at maximum. The generators operate at 360 r.p.m., the high-pressure turbine at 4,000 and the low-pressure at 3,000 r.p.m. Steam at 375 lb. 700 deg. F. is applied, the low-pressure units exhausting at 28½-in. vacuum. Arrangements are made for bleeding steam from the low-pressure cylinder.

The construction, as indicated in Fig. 1, inherently posesses several points of superiority over that of a single-cylinder geared unit. Pinions placed on opposite sides of the main gear, tend to neutralize side thrust in the low-speed member. The width of gear face is approximately one-half that required, as compared with a gear having only one driving pinion.

Reliability may be greatly increased. Abnormal conditions which would incapacitate one of the turbines, may be met by the remaining element so as to deliver approximately three-quarters normal load. This is considerably better than a total shutdown, which would occur under similar circumstances in a single-cylinder unit. Operating conditions in this particular instance are such that it is not necessary to provide for this full available flexibility. Additional piping and other special equipment for independent operation of the low-pressure turbine are not here included.

By arranging the high- and low-pressure turbines to operate at speeds most suitable to design requirements, each element may be built most favorably with regard to efficiency and material.

The design of the unit centers around that of the reduction gear. The speed of the high-pressure turbine is limited by dimensions of the pinion as will be shown hereinafter. It is therefore desirable to arrange as high a speed as is consistent with conservative design, as the tendency is to increase efficiency and diminish the material of the turbine as the speed is increased. The blading material decreases approximately as the square of the speed.

The speed of the main gear is limited to that of the direct-current generators that it must drive; this is 360 r.p.m. The maximum diameter of this gear is therefore determined from considerations of conservative tooth-speed limits. The pinion of the high-pressure turbine is then proportioned for the smallest reasonable diameter which, in this case, gives 4,000 r.p.m. A high-pressure turbine of this capacity without limitation as to pinion speed could be built at approximately 10,000 r.p.m. for best commercial results. The pinion for the low-pressure turbine is not required to be of minimum diameter. It is proportioned for the most desirable speed with regard to the design of this element, 3,000 r.p.m., and therefore is larger than the high-pressure pinion.

Two Pinions Reduce Gear Face Width

The two turbines must give equal torque at about normal load. With one pinion on each side of the main gear, as shown in Fig. 1, it will be seen that only one-half of the width of face on the main gear is required as compared with that which would be necessary if only one driving pinion were used. With two pinions, pres-

sure is applied on two sides so that twice the amount of energy at a given tooth pressure may be transmitted, as compared to only one pinion.

Reduction gears in general fall into two classes. In the first there is a certain amount of flexibility so that the pinion is able to align itself to some extent with the gear in order to compensate for irregularities. In the second class of reduction gears irregularities of all kinds are kept within certain desired limits, and they are so constructed that all bearings are absolutely located and rigidly maintained in their places. This turbine gear is of the latter type.

The pinions are composed of solid nickel-steel forgings, and the main gear of a rolled seamless steel ring shrunk over a cast-iron center. This insures uniform metal of the proper consistency from which the gear teeth are cut. The helical angle is 45 deg., which makes somewhat more difficult cutting, but is believed to give better results than smaller angles. Low tooth pressure is of prime importance.

A steady supply of oil is provided from an overhead tank. A number of small nozzles project oil in a continuous sheet, against the teeth as they run together.

Reduction Gear Levelled Up First

The reduction gear is regarded as the center of the set. When being erected, the gear casing is first leveled up on the foundations. The other elements are then lined up in accordance with the coupling flanges. Flexible couplings are in all cases supplied. The generator is connected to the slow-speed gear shaft by the standard pin and rubber-bushing type coupling. The turbines are connected to their respective pinions by the "Fast" type of flexible coupling in which the contact surfaces are of gear-teeth design and operate in a bath of oil.

Both turbines are of the general girder construction, as in Fig. 2, being fastened rigidly to the gear case and allowed to expand away from the gear case by means of a sliding fit on the sole plates which support the outboard bearings. The lower elements, or sole plates proper, are rigidly fastened in place. The turbine pillow-blocks are held to the sole plates by means of shouldered bolts. Lubricated machined guide strips determine the location sideways.

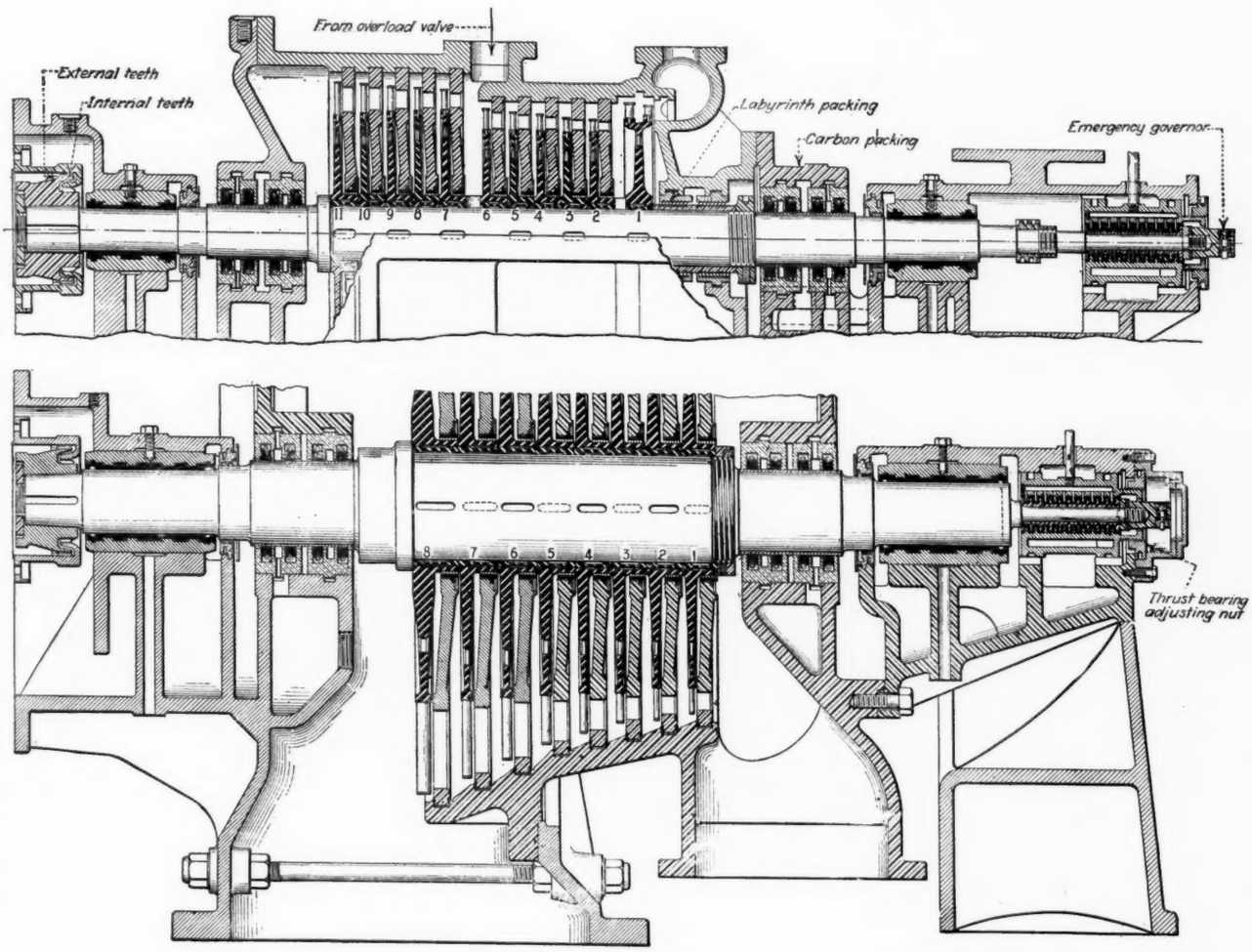
From overload valve
External teeth
Internal teeth
Labyrinth packing
Carbon packing
Emergency governor
11 10 9 8 7 6 5 4 3 2 1
8 7 6 5 4 3 2 1
Thrust bearing adjusting nut

Fig. 2—A steel casing with no internal bolts contains the high-pressure rotor, above. Low-pressure rotor below

The high-pressure turbine, shown at the top in Fig. 2, contains one double-row wheel and ten single-row wheels. Steam enters at 375 lb. gage and is discharged from the first stage at approximately 125 lb. absolute. This turbine exhausts at approximately 25 lb. absolute at full load.

Steam is admitted by a primary and a secondary valve to two sets of nozzles, shown in Fig. 3. It will be seen that practically full peripheral admission is thus obtained with both valves open, which is approxi-

mately full load. Steam entering through the secondary or upper valve may also be admitted to an overload valve, which is placed so as to admit steam to the seventh stage of the turbine. Arrangements are made for bleeding steam from the seventh stage and from the connecting pipe from the high- to the low-pressure turbine. The bled steam is controlled by a specially designed valve.

No Internal Bolts in Steel Casing

The high-pressure turbine is remarkable from the fact that no internal bolts are utilized within the high-pressure casing. The casing and diaphragms are all of steel so as to be suitable for high pressures and temperatures. The diaphragm packing containers are riveted. The high-pressure nozzle plates are held in place by calking strips, which in turn are welded over. This makes the nozzle plates integral with the cylinder, and yet allows them to be renewed when necessary.

Turbine wheels are of special carbon steel pressed over split tapered steel bushings. Keys on the shaft extend through the bushings so that they also hold the wheels. It will be seen in Fig. 2 that a collar is machined on the low-pressure end of each shaft against which the wheels are assembled. The wheels are pressed toward this shoulder over the tapered bushings one at a time, by means of a nut operating on the thread at the high-pressure end of each turbine shaft. Suitable loose sleeves must be provided to convey the pressure from the nut to the wheel. The tightness of the fit of the wheel can thus be accurately controlled.

The bushings are recessed in the region between each end, so that the wheel really contacts on two surfaces, one being directly under the web and the other at the end of the shoulder. Although centrifugal force or other factors may tend to diminish the pressure under the web with which the wheel is fixed, there will be practically no forces acting, comparatively speaking, on the collar so that this, under all circumstnces, will maintain a tight fit.

Static Balance Sufficient

All wheels are carefully balanced on specially made arbors, before being bucketed. After bucketing, the wheels are rebalanced before finally being mounted on the shafts. This static balancing of the wheels is carried out so accurately that no further balancing is necessary after the wheels are mounted on their shafts.

Diaphragms are solid and the casings are split. The diaphragms are built by pinning nickel-bronze vanes in center disks and shrinking a ring around the outside of the vanes.

The high-pressure end of the high-pressure turbine contains both labyrinth and carbon packing. The labyrinth, which is next to the wheel, consists of vanes brought closely to the packing sleeve. Steam leaking between this and the carbon packing is piped to the seventh stage of the turbine, while that from the region between the two labyrinth rings is taken to the fifth stage. Steam leaking by the first two rings of carbon packing goes to the fourth stage of the low-pressure cylinder.

Carbon rings of the standard construction are held in place by garter springs and also flat springs for maintaining the proper position. Rings are prevented from turning by a keyed member. They are all arranged to be disassembled if necessary without removing the turbine casing. Rings for the labyrinth packing are free to move in position in accordance with shaft movement and are supported by spring pressure.

Labyrinth packing glands in the diaphragms, however, are held in place by a certain degree of friction. They are free to move in accordance with the changing shaft position and are maintained in the adjusted position by the friction thus supplied. The glands are made of monel metal and brass with the usual fin construction.

The standard type of centrifugal governor is mounted on the high-pressure cylinder and geared to the shaft, the worm being shown in Fig. 2. Primary and secondary valves are controlled through a hydraulic valve gear. The primary valve is opened first by the hydraulic cylinder and is indicated in Fig. 3. After this has opened the predetermined amount, a collar on the shaft shown at *L*, contacts with the secondary valve, thereupon lifting it. Additional travel of the hydraulic cylinder will open the overload valve. Hand valves are supplied in each steam chest for regulating nozzle pressure so as

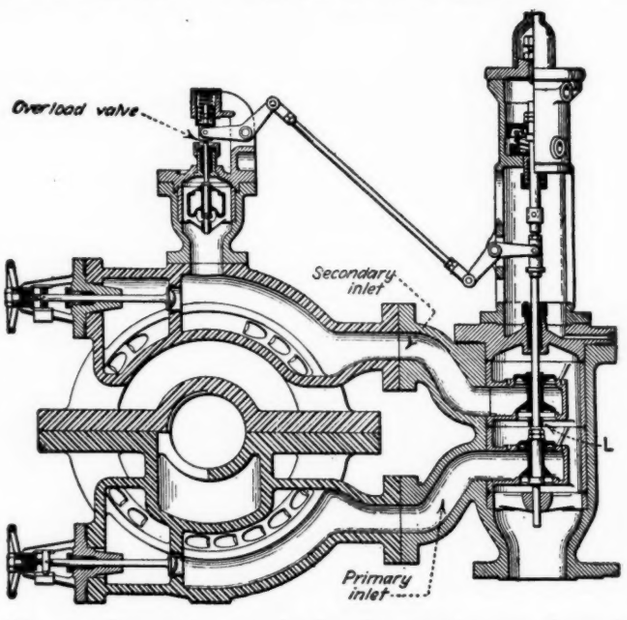

Fig. 3—Steam for overload valve must pass through secondary valve

to give a more efficient distribution at various loads than that supplied by the governing valves alone. The hand valves are adjusted so as to give as near live steam pressure in the steam chests as possible. An emergency governor is contained in the extreme end of the shaft as indicated in Fig. 2.

Bearings Recessed at Each Side

Bearings are of the usual type, being horizontally split with cast-iron shells, babbitt lined. They are not provided with oil grooves, but are recessed on each side. Oil is fed from both sides at about 5 lb. pressure and approximately 100 deg. F. This is supplied at constant pressure from an overhead tank. Oil is cooled as it goes into the tank after being pumped from the turbine. The thrust bearings are of the usual marine type. The adjusting nut is placed at the end and fastened by a setscrew. It is inclosed by a cast-iron cover, except when being adjusted.

The steam consumption of this turbine set when operating with 375 lb., 700 deg. F. steam and exhausting into a vacuum not less than 28½ in., is guaranteed not to exceed 11.5 per kw. at full load of 4,000 kw.

Reconnecting Direct-Current Machines —Changing Speed and Voltage

Problems Worked Out Showing How To Figure a New Winding for a Shunt, Compound and a Series Motor To Operate With a Change in Speed and Voltage

By A. C. ROE
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ALTHOUGH the most frequent change that has to be made in motor windings is a change in voltage, it sometimes occurs that conditions have to be met where a change in voltage combined with a change in speed have to be taken care of. Such conditions generally require a new winding on the armature, since even though the change in voltage can be accommodated by regrouping the armature coils, the change in speed will generally be of such a value as to require a new set of coils. In working out such problems, the simplest way is to consider first a winding for a change in voltage and then change this winding for the new speed.

Consider the case of a shunt motor with interpoles, the original rating being 5 hp. 220 volts 20 amperes

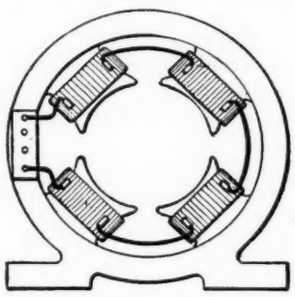


Fig. 1—Field coils connected series

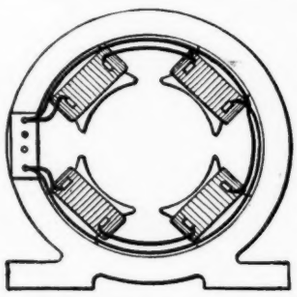


Fig. 2—Field coils connected series-parallel

850 r.p.m., with four main poles and two interpoles. Each shunt coil has 1,920 turns of one No. 24 s.-c.-c. wire, each of the two interpole coils has 122 turns of No. 8 d.-c.-c. square wire arranged in four layers of 19 turns, one of 15, one of 11, one of 8, one of 7 and one of 5 turns, a total of nine layers. The armature has 33 slot and 99 commutator bars, each armature coil consisting of six turns of one No. 13 d.-c.-c. wire. The complete coil is made up of three coils taped together so that it is three conductors wide by six deep. The coils are connected into a wave winding with a coil pitch 1 and 9, and a lead pitch of 1 and 51. A brush rigging is used having four brush arms, one brush per arm, $\frac{3}{8}$x1$\frac{1}{2}$ inches.

For the original winding the current density per square inch of brush in contact with the commutator equals the area of the brushes of one polarity divided into the full-load current. There are two brushes of one polarity, so that the brush cross-section area is $1\frac{1}{2} \times \frac{3}{8} \times 2 = 1\frac{1}{8}$ sq.in. and the current density = 20 $\div$ $1\frac{1}{8}$ = 18 amperes per square inch approximately. It is desired to change this motor to operate on 110 volts at 1,260 r.p.m. Considering the shunt-field coils first, which are connected in series for 220 volts, as in Fig. 1, they can be connected series-parallel for 110-volt operation, that is, two groups two coils in series and the two groups connected in per parallel, as in Fig. 2. The armature can be changed for 110 volts by changing from a wave to a lap winding or by reducing the turns per coil one-half and doubling the cross-section of copper and retaining the wave winding. The latter method will be tried first. The coils can be wound with three turns of two No. 13 d.-c.-c. wires in parallel, using the original lead and coil pitch when grouping the coils in the winding. Since the number of turns in the armature coil vary inversely as the speed, the turns per coil for the new speed, or 1,260 r.p.m., will equal (3 $\times$ 850) $\div$ 1,260 = 2.02, or two turns per coil. The ampere-turns for the 220-volt 860-r.p.m. coils were (20 $\div$ 2) $\times$ 6 = 60. For the same ampere-turns in the 110-volt 1,260-r.p.m. two-turn coils, the current in each armature circuit would be 60 $\div$ 2 = 30, which is three times the original current per circuit and three No. 13 wires in parallel will have the proper size copper. The 110-volt 1,260-r.p.m. coil will therefore be wound with two turns of three No. 13 d.-c.-c. wires per coil.

Effect of Changing to Lap Winding

By changing to a lap winding, the six turns coil will be good for 110-volt operation. Then, turns per coil for 1,260 r.p.m. equals (6 $\times$ 850) $\div$ 1,260 = 4. Although this number of turns can be used, it will require increasing the size of the conductors in proportion to the speed if the horsepower is to be increased. Increasing the size of the conductors will make it difficult to accommodate the new winding to the slots, therefore in this case the wave winding would make the better job. As the interpoles are in series with the armature, the same current will flow in both. The original ampere-turns per interpole coil was 20 $\times$ 122 = 2,440. At the 110-volt and 1,260-r.p.m. rating the full-load current is 60 amperes. Then, for 2,440 ampere-turns and 60 amperes the turns required will be 2,440 $\div$ 60 = 40.7, or 40 turns per coil will do. The current has been increased three times, therefore the size of wire will have to be increased in the same proportion or the coils can be wound with three No. 8 square wires in parallel, wind four layers of 6 turns, 1 of 5, 1 of 4, 1 of 3, and 2 of 2 turns per layer. The 1,260-r.p.m. horsepower rating will be (5 $\times$ 1,260) $\div$ 850 = 7.4. The rating of the motor would therefore be $7\frac{1}{2}$ hp., 110 volts, 1,260 r.p.m. and 60 amperes. A brush would have to be selected that would permit a current density of 60 $\div$ $1\frac{1}{8}$ = 53 amperes per square inch. If such a brush cannot be obtained to give satisfactory operation, then new brush-holders will have to be provided that will permit the use of larger brushes.

A 50-hp. compound-wound 110-volt 900-r.p.m. motor

is to be changed for operation on 220 volts at 600 r.p.m. The full-load current is 375 amperes and the motor has four poles. Each shunt coil consists of 952 turns of one No. 13 s.-c.-c. wire arranged in 32 layers; 16 layers of 32 turns, 2 of 31, 2 of 30, 2 of 29, 2 of 28, 2 of 27, 2 of 26, 2 of 25, and 2 of 24 turns. Each series coil has 3½ turns of one ⅛x2½-in. bare copper strap, wound on edge. The armature is wave wound, two circuit, having 55 slots, and commutator bars. The coils have one turn of copper strap ⅜x⅜ in., coil pitch is 1 and 13 and the lead pitch is 1 and 29.

The first step is to check the voltage per bar to determine whether the same 55-bar commutator can be used for the higher voltage. The volts between bars for the 110-volt winding was 110 ÷ (55 ÷ 4) = 8 volts, and for the 220-volt winding the voltage between bars will be doubled or equal 8 × 2 = 16 volts, which is a safe value, therefore the 55-bar commutator can be used for the 220-volt winding. Since the voltage is doubled, the shunt-field coils will have to be rewound with 952 × 2 = 1,904 turns of wire having one-half the cross-section in the original winding. A No. 16 wire has one-half the cross-sectional area of a No. 13, therefore this size either single-cotton or enameled covered, can be used. It will probably be found best to use enameled-covered wire on account of the smaller wire requiring more space than the large wire when wound into a coil. At 900 r.p.m., the original speed, a two-turn armature coil of half the cross-section of the 110-volt winding will be required for 220 volts. Then, at 600 r.p.m., the turns per coil for 220 volts will be (2 × 900) ÷ 600 = 3. The number of turns in the armature coils have been increased three times, therefore the cross-section of copper will have to be reduced one-third. Dividing the original strap into three sections will give a conductor of ⅜x⅜ in., or ⅜-in. square. A ⅜ = 0.1875 in. and the nearest diameter of square wire to this is a No. 5, which has a diameter of 0.182 in. bare, as indicated in the table. The original conductor has an area of 0.103582 sq.in., and the new size should be 0.103582 ÷ 3 = 0.034527 sq.in. A No. 5 square wire has an area of 0.03120 sq.in., which is close enough. Therefore the 220-volt 600-r.p.m. armature winding will consist of 55 coils three turns per coil of one No. 5 sq. d.-c.-c. wire, one wide by three deep per coil. The commutator necks will have to be filled up with four No. 5 square wires. The coil pitch will remain 1 and 13 and the lead pitch 1 and 29 as in the original winding. The size of the copper has been decreased by one-third, therefore the armature current will have to be decreased by this value, or 375 ÷ 3 = 125 amperes will be the full-load current for the new winding.

Since the current has been decreased by 3, the number of turns in the series coil will have to be increased by 3 and wire one-third the original size used. The original coils had 3.5 turns; then the new coils will have 3.5 × 3 = 10.5 turns per coil. In the series coils the original conductors were ⅛x2½ in. which has an area of 0.309146 sq.in. Reducing either dimension of the original conductor by 3 will still require copper of a size that is hard for the general repair shop to wind. Therefore the next step would be to reduce the area in square inches to circular mils and check the winding

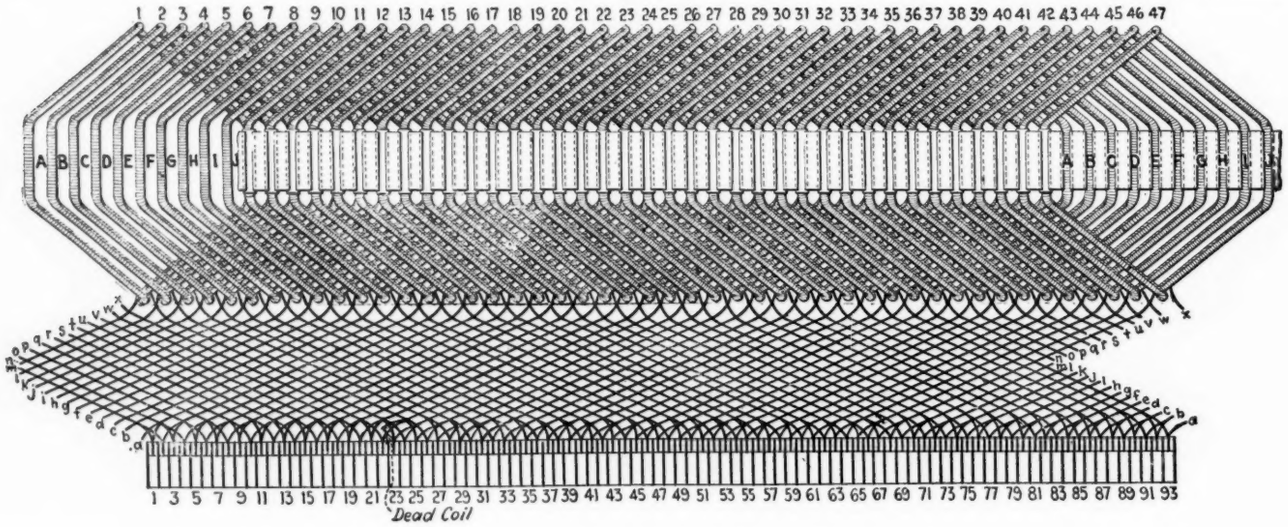


Fig. 3—Four-pole wave winding having 93 coils

space per coil to determine the prospects of substituting a number of round or square wires in parallel. A circular mil equals one square inch ÷ 0.0000007854. The new conductor is to have a cross-sectional area equal to one-third that of the original, or 0.309146 ÷ 3 = 0.103049 sq.in. Then the new size copper in circular mils equals 0.10304 ÷ 0.0000007854 = 131,194 circ.mils. Dividing by 2 and 3 we get 131,194 ÷ 2 = 65,597 circ.mils for two wires in hand and 131,194 ÷ 3 = 43,731 circ.mils for three wires in hand. From the wire table it will be found that a No. 2 round wire has an area of 66,370 circ.mils and a No. 4 an area of 41,740 circ.mils. Checking the available winding space for the series coils, it is found to be 1½x3 in. A No. 2 d.-c.-c. wire has an insulated diameter of 0.276 in. or six No. 2 wires would require 0.276 × 6 = 1.656 in. A No. 4 d.-c.-c. wire has an insulated diameter of 0.222, then nine No. 4 wires would require 0.222 × 9 = 1.998 in. A No. 3 square wire has a diameter of 0.252 in. and an area of 62,732 circ.mils; six wires would require 6 × 0.252 = 1.512 in. Two of these wires in parallel give approximately the required cross-sectional area, consequently six wires per layer are equivalent to three turns. The new coils are to have 10.5 turns, and with three turns per layer they will have 10.5 ÷ 3 = 3.5 or 4 layers, which is equivalent to a depth

of 4 × 0.252 = 1.008 in. This dimension is such as to come well within the space limitations and allow for spacers between the layers to improve the ventilation of the coils. Therefore the new coils will be wound with 10.5 turns of two No. 3 square wires in parallel three turns per layer for three layers and the fourth layer will have 1½ turns. Since the horsepower rating of the motor varies with the speed, the new rating at the reduced speed would be (50 × 600) ÷ 900 = 33⅓ hp. Owing to the reduced speed the ventilation would be poorer, therefore a safe rating would be about 30 hp., 220 volts, 600 r.p.m. and about 115 amperes.

A 15-hp. 220-volt 500-r.p.m. 4-pole series motor is to be rewound for 110 volts and a speed of 700 r.p.m. The 15-hp. 220-volt 500-r.p.m. winding data are 4 series coils each consisting of 52 turns of one No. 2 d.-c.-c. wire, arranged in four layers of 13 turns per layer. The armature has 47 slots, 93 commutator bars, one dead coil, wave wound, Fig. 3. Each armature coil has

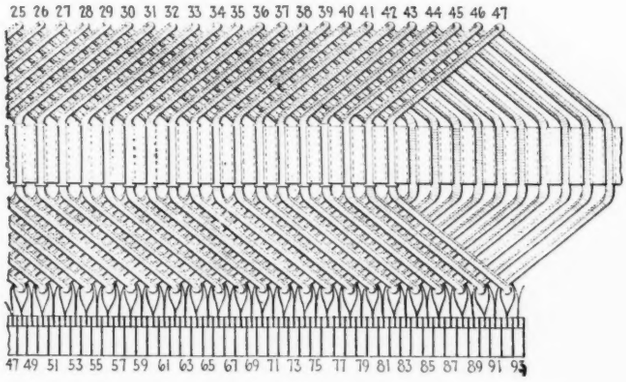


Fig. 4—Part of winding Fig. 3 grouped lap

three turns of two No. 12 d.-c.-c. wires in parallel. The coils are grouped in pairs and taped together, so that there is a total of 47 coils. The coil pitch is 1 and 11 and the lead pitch 1 and 48. There are four brush-holders, each containing one brush ½x2⅜ in. Then the current density per brush equals 58 ÷ (2 × ½ × 2⅜) = 24.4 amperes per square inch. First considering the armature circuit, the voltage change can be made satisfactory by reconnecting the original winding for a lap grouping of the coils, as shown for part of the winding, Fig. 4. The change in armature-coil turns required for the new speed is (3 × 500) ÷ 700 = 2.14, or two turns per coil. Using two turns per coil will increase the speed to (3 × 500) ÷ 2 = 750 r.p.m. and the hp. to (15 × 750) ÷ 500 = 22½. The armature ampere-turns per coil for the 15-hp. 220-volt rating was (58 ÷ 2) × 3 = 87; then with two turns per coil, the current per coil or path for the 110-volt, 750-r.p.m. winding will have to be 87 ÷ 2 = 43.5 amperes, and the full-load rating will be 43.5 × 4 = 174 amperes. Checking the current density per brush gives 174 ÷ (2 × ½ × 2⅜) 73.2 amperes per square inch, which is too high so that larger brushes will have to be used. A brush having an allowance of 45 amperes per square inch will require a cross-sectional area of 174 ÷ (2 × 45) equals 1.93 sq.in. Using the same length, the new thickness will be 1.93 ÷ 2⅜ = 0.8 in. or a ¾-in. brush. This brush is probably too thick for satisfactory operation on this size machine. Unless the width of the brush can be increased so that the thickness can be kept to less than ¾ of an inch, it may be necessary to reduce the capacity of the machine.

In the series-field coils the old winding had 58 × 52 = 3,016 ampere-turns per coil. The new ampere rating is 174 and turns per coil will have to be 3,016 ÷ 174 = 17.3 turns, or 17 turns per coil. One No. 2 round wire has an area of 66,370 circ.mils and 66,370 ÷ 58 = 1,144 circ.mils per ampere. For 174 amperes there will be required 1,144 × 174 = 199,056 circ.mils, or three No. 2 wires in parallel will do as they have 3 × 66,370 = 199,110 circ.mils total cross-sectional area. The coils can be wound with 17 turns of three No. 2 d.-c.-c. wires in parallel, four turns per layer for five layers, the fifth layer having two turns.

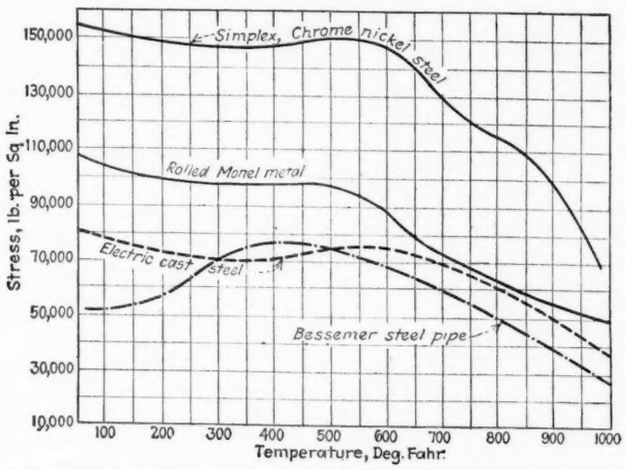
PROPERTIES OF SQUARE COPPER WIRE

Size B. & S. Gage	Diameter in Inches—Bare	D.-C.C.*	T.-C.C.	Area, Circ. Mils	Area, Sq.In.
0000	0.460	0.483	0.497	265,152	0.20825
000	0.410	0.433	0.442	209,345	0.16442
00	0.365	0.388	0.397	165,037	0.12972
0	0.325	0.348	0.357	130,124	0.10220
1	0.289	0.312	0.321	102,292	0.08034
2	0.258	0.281	0.290	80,203	0.06300
3	0.229	0.252	0.261	62,732	0.04927
4	0.204	0.227	0.236	50,737	0.03985
5	0.182	0.205	0.214	39,725	0.03120
6	0.162	0.183	0.191	32,340	0.02540
7	0.144	0.163	0.171	25,439	0.01998
8	0.129	0.146	0.153	19,951	0.01567
9	0.114	0.129	0.135	16,221	0.01274
10	0.102	0.115	0.120	12,783	0.01004
11	0.0907	0.103	0.108	10,033	0.00788
12	0.0808	0.093	0.098	7,868	0.00618
13	0.720	0.083	0.088	6,162	0.00484

* D.-C.C., double-cotton covered; T.-C.C., triple-cotton covered.

Ultimate Strength of Monel Metal

Attention has been called to the curves showing Elastic Limits and Ultimate Strengths of Metals at High Temperature, which formed part of the paper by W. S. Monroe on "High Pressures and High Temperatures in Central Stations," read at the World Power Conference and abstracted in the July 15 issue. In



Ultimate strength of metals at high temperatures

these curves the elastic limit of monel metal appears at some points greater than the ultimate strength.

This was taken up with the office of the author, which advises that, after carefully checking back the source of the data, it was discovered that through an oversight the curves were copied from two separate tests, the "Ultimate Strength Curve" being for cast monel metal and the "Elastic Limit Curve" for rolled monel metal. The corrected set of curves for ultimate strength, covering rolled monel metal, is reproduced herewith. This replaces Fig. 3 of the original article.

Semi-Diesel Operation at All Load Ratings

By H. F. SHEPHERD

IT HAS been known that certain types of the surface-ignition, or semi-Diesel, oil engines possess variable operating characteristics through the load range. If steady at light loads, they might be prone to overheat at rating, or if possessed of a good overload capacity and ready heat dissipation at rating, they might be unstable

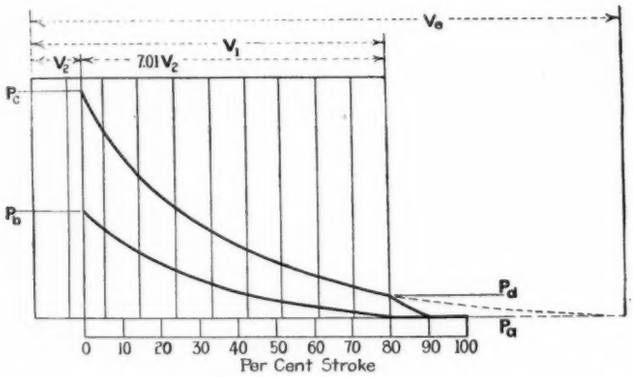


Fig. 1—Diagram for two-cycle surface-ignition engine

in operation at moderate loads, unless compensating devices are fitted.

Several influences enter into the difficulty, the most important of which is scavenging. Let us choose as an example a two-stroke-cycle engine having a compression of 180 lb. gage, and operating at such an elevation that the initial or atmospheric pressure is 14 lb. absolute. Assume a compression coefficient of $n = 1.35$ and for expansion of $n = 1.2$, values common to the more primitive solid-injection engines; also let:

V_1 = Initial cylinder volume at point where piston covers exhaust ports = 100;
V_2 = Clearance volume;
P_a = Initial pressure = 14 lb. per sq.in.;
P_c = Compression pressure = 180 lb. per sq.in. gage.

The value of V_2 may be found from the equation for compression; using a value of n of 1.35, as follows:

$$\frac{P_b}{P_a} = \left(\frac{V_1}{V_2}\right)^{1.35} \quad \text{or} \quad \frac{180 + 14.7}{14} = \left(\frac{100}{V_2}\right)^{1.35}$$

$$V_2 = 14.26$$

For expansion, since $\left(\frac{V_1}{V_2}\right)^{1.2} = \frac{P_c}{P_d}$, we may substitute 7.01 for the volume ratios, making the pressure ratio $\frac{P_c}{P_d} = 10.35$.

These relations established, diagrams for various mean effective pressure values may be constructed. A sketch of the proposed diagram is shown in Figure 1.

The first step is to learn the absolute mean pressure P_{ab} of the compression line. The work formula for the area under the curve $P_a\ P_b$ is

$$W = \frac{P_b\ V_2 - P_a V_1}{n - 1}$$

in which W is in foot-pounds, pressures in pounds per square foot, and volumes in cubic feet, and n equal to 1.35. By substitution we have

$$P_b V_2 - P_a V_1 = P_{ab}\ (V_1 - V_2)$$

$$Pa_b = \frac{P_b V_2 - P_a V_1}{0.35\ (V_1 - V_2)} = 45.53\ \textit{lb. per sq.in.}$$

We must next establish the relation between P_c and P_d and the mean pressure P_{cd} of the expansion line. By the same method used in finding the compression pressure, we write the expression for the expansion m.e.p. as:

$$P_{cd} = \frac{P_c V_2 - P_d V_1}{(n-1)\ (V_1 - V_2)}$$

$$P_c = 10.35\ P_d$$

$$n - 1 = 0.2$$

By substitution of the proper values we have

$$P_{cd} = \frac{(14.26\ P_c) - (0.0966\ P_c \times 100) = 0.2683\ P_c}{0.20\ (85.74)}$$

$$P_{cd} - P_{ab} = \textit{m.e.p. of diagram}$$

$$m.e.p. = 0.2683\ P_c - 45.53$$

$$P_c = \frac{m.e.p. + 45.53}{0.2683}$$

$$P_d = \frac{m.e.p. + 45.53}{2.7761}$$

Having established the dimension of a diagram for any mean effective pressure, we next examine into the temperatures during the cycle, calling T_a the charge temperature, T_b the compression temperature, T_c the explosion temperature, T_d the expansion-end temperature and T_e the temperature of the gases after expanding adiabatically through the exhaust ports. To find T_e, we must find the volume V_e after expansion through the exhaust ports. For adiabatic expansion of air $P_e V_e^{1.41} = P_d V_1^{1.41}$, or

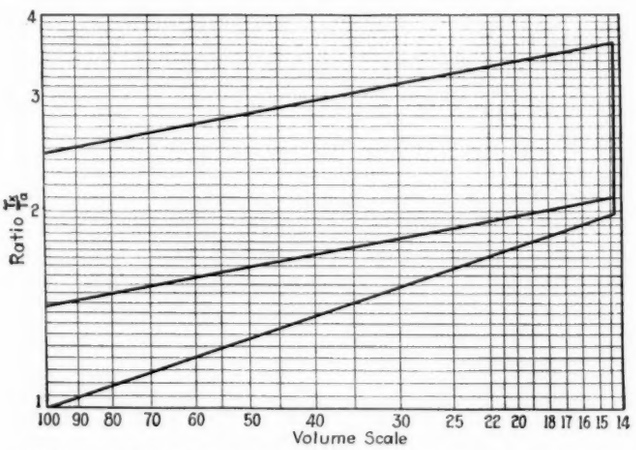


Fig. 2—Relation of temperature ratio and cylinder volume

$$V_e = V_1 \sqrt[1.41]{\frac{P_d}{14}} \qquad T_e = \frac{V_e}{V_1}\ T_a$$

T_a is variable, depending largely on the degree of scavenging and also on the heating that the charge

receives during introduction through contact with the walls. For this reason it is best at this stage to deal with temperatures as ratios, making $T_a = 1$; and T_x and T_y, the temperatures at any point on the compression or expansion lines, respectively, are then expressed in units of T_a degrees by numerical values equal to $\frac{T_x}{T_a}$ or $\frac{T_y}{T_a}$.

First computing the various end points on the card at probable mean effective pressures, we have:

M.e.p.	P_c	P_d	V_e
10	206.97	20.00	1.287
20	244.24	23.61	1.4485
30	281.51	27.21	1.6020
40	318.78	30.81	1.7496
50	356.06	34.41	1.8924
60	393.33	38.01	2.0308

From the general relation $\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$ and the foregoing data, taking $T_a = 1$ we may compute ratios $\frac{T_x}{T_a}$ and $\frac{T_y}{T_a}$ for the other fixed points, obtaining the following value:

M.e.p.	$\frac{T_c}{T_a}$	$\frac{T_d}{T_a}$	$\frac{T_e}{T_a}$
10	2.1089	1.4187	1.287
20	2.4893	1.6864	1.4485
30	2.8690	1.9436	1.6020
40	3.2485	2.2007	1.7496
50	3.6280	2.4578	1.8924
60	4.0076	2.7150	2.0208

$\frac{T_b}{T_a}$ may be ascertained as follows:

$$\frac{T_b}{T_a} = \frac{P_bV_2}{P_aV_1} = \frac{194 \times 14.26}{14 \times 100} = 1.976$$

These ratios may now be plotted on logarithmic cross-section paper as in Fig. 2, but as we are concerned in the hot-surface engine with a problem of radiation, the case must be studied on a temperature-time basis and not on a temperature-volume or stroke-fraction basis.

With exhaust ports 20 per cent of the stroke in length and connecting-rod angularity neglected, the following relations exist between the volume V_x in fractions of V_1 and the crank angles in degrees, and a rod to crank ratio of $4\frac{1}{2}$:

V_x	Crank Angle, Deg.	V_x	Crank Angle, Deg.
0.1426	0	0.30	45.05
0.16	14.58	0.40	58.68
0.17	18.38	0.50	70.53
0.18	21.50	0.60	81.58
0.19	24.25	0.70	92.30
0.20	26.75	0.80	103.16
0.21	29.03	0.90	114.42
0.22	31.15	1.00	126.87
0.25	36.88		

Using this table in conjunction with the logarithmic sheet Fig. 2, a time-temperature curve is established for both the constant compression line and the expansion line for any desired mean effective pressure. Curves for 10 lb. and 50 lb. m.e.p. on a time-temperature basis are shown in Fig. 3. From these by means of a planimeter one finds that with 10 lb. m.e.p. the mean cyclical temperature ratio $\frac{T_m}{T_a}$ is 1.45, while with a mean effective pressure of 50 lb., the ratio is 1.99.

From these values one may conclude that 10 lb. m.e.p. based on the effective stroke, is not far from the idling m.e.p. for a two-stroke-cycle hot-bulb engine. Assuming perfect scavenging and a temperature T_a of 520 deg. abs., $T_m = 1.45\ T_a$ or $1.45 \times 520 = 754$ deg. abs., or 294 deg. F. A hot bulb must remain at 600 deg. for even reasonably good working, and it should be somewhat hotter. We certainly cannot expect to keep a surface at 600 deg. when it is exposed to a mean temperature of only 294 deg. We know that in certain types it does keep hot, so the initial temperature must be in the neighborhood of $\frac{600 + 460}{145} = 731$ deg. abs., or 271 deg. F. This is accomplished either deliberately or accidentally, by retaining the products of the previous combustion in each new charge. The relative quantity of hot neutrals retained may be determined, theoretically, as follows: For 10 lb. m.e.p. $T_c = 1.287\ T_a = 1.287 \times 731 =$ 941 deg. abs., or 481 deg. F.

It is now proposed to mix fresh air at 60 deg. F. with exhaust gas at 481 deg. to produce a mixture of the two at 271 deg. We may consider these two gases as measured separately and allowed to mix. The fresh air undergoes a rise of temperature $271 - 60 = 211$ deg. F., the exhaust gas constituent of the charge is reduced in temperature $481 - 271 = 210$ deg. F. The specific heats being the same and assuming a charge in which

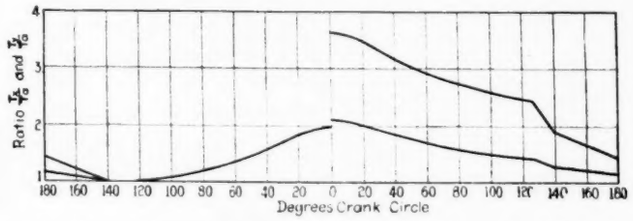


Fig. 3——Temperature ratio and crank-angle relation

X equals the relative weight of fresh air and Y equals the relative weight of residuals, we have $211\,X = 210\,Y$ and $X + Y = 1$. The ratio $\frac{X}{Y}$ becomes 0.995, or $X = Y$ for all practical purposes.

If the charge equals unit weight at 520 deg. abs., the weight of air present reduced to 60 deg. F., or 520 deg. abs., is $\frac{520}{731} \times 0.50 = 0.36$. In the final working charge of the design the relative weights of air and residuals are as 50 to 50 when working at 10 lb. m.e.p. The relative weight of air to that properly contained in V_1 at 60 deg. F. is 36 to 100. Thus the engine may not be better scavenged than 36 per cent, considering a cylinder full of air at 60 deg. as 100 per cent scavenged. Continuing on the bases of T_a equals 271 deg., or 731 deg., to insure light-load operation T_b is equal to 1.976 and T_a becomes = 1,444 deg. abs., or 984 deg. F., a per missible figure, as the fuel is all in before dead center.

We have satisfied this condition for friction load, but let us advance to 50 lb. m.e.p. based on the effective stroke, or approximately the rating for the type. A cylinder charge at 60 deg. F. equals unit weight. Then at T_a the charge weight equals $\frac{520}{T_a}$. By proposition the air charge weight equals 0.36, also by the table T_c 1.8924 T_a. Then the weight of the residuals in the charge equals $\frac{520}{T_a} - 0.36$.

Upon mixing we have $0.36\ (T_a - 250) = \left(\frac{520}{T_a} - 0.36\right)\ (1.8924\ T_a - T_a)$ or $0.36\,T_a - 187.2 = 464 - 0.32\ T_a$, or $T_a = 958$ deg. ab., or 498 deg. F.

There is no need to multiply this value of T_a by the ratios to show that it will result in a compression-end temperature so high as to cause cracking of the injected

fuel, and a mean temperature so high as to throw excessive heat to the hot surface and to cause piston heating. The condition is an impractical one for the so-called "dry" engine. It accounts for the copious drafts of injection water required by some engines which perhaps idle most perfectly.

What is required is that T_a equal approximately 731 deg. F. abs., as it did under the 10 pounds. Let $Y =$ relative weight air and Y = relative weight of neutrals. Then

$$X(T_a - 520) = Y(1.8924\,T_a - T_a)$$
$$X(731 - 520) = Y(1.8924 \times 731) - 731$$
$$211\,X = 652\,Y$$
$$X + Y = 1$$
$$X = 3.09\,Y$$
$$Y = 0.244$$
$$X = 0.756$$

The air charge weight equals $\frac{520}{731} \times 0.756 = 0.54$ units when $V_1 = 1$, or unit weight of air at 60 deg. F.

These figures are attainable even with a 1-to-1 scavenge-pump ratio, but it is clear that with an engine of the type and performance proposed in the first paragraph, any condition that insures satisfactory operation at friction load will cause overheating at rating, and any condition that is right for rating will not provide sufficient heat for the hot surface at friction load.

In practice there are several courses for the correction of this state of affairs, most of which amount to setting a thief to catch a thief. These are:

1. Increase the scavenging efficiency as the load increases.

2. Use a constant low-scavenging efficiency to insure light-load operation and use water injection to cool the charge at heavier loads.

3. Use a constant high efficiency of scavenging and some one of the many patented compensating devices to be thrown in at light loads or idling to cause the oil injecting to be early and so permitting a great heat absorption by the hot surface.

4. Combinations of any two or three of the foregoing.

A far better suggestion is offered and practice is working around to it slowly but surely: Scavenge as nearly 100 per cent as possible and choose a compression high enough to give a proper compression-end temperature which will be fairly constant with increasing load if the metal parts are cooled well. This means larger pumps on all two-stroke-cycle engines now employing the 1-to-1 system and careful attention to exhaust systems for all solid-injection engines.

With 100 per cent scavenging there can be no variation of T_a or T_b theoretically. Of course this ideal is impossible, but the engine that approaches nearest equable working at all loads will scavenge to the practical limit. It will be developed for the least possible after-burning, thus reducing T_a and T_c since low-temperature residuals will have the least effect upon T_a. It must be a moderately high compression engine, since the desired value of T_b must be reached by compression and not by raising T_a through admixture of hot residuals.

Present Practice in Steam Generation in the United States*

By Dr. D. S. Jacobus†

THERE are a number of factors to be considered in determining the economy of a power plant. The economy obtained through heating the feed water by means of steam bled from the steam turbine is one of these. Where economizers are used, heating the feed water by steam extracted from the turbine reduces the heat absorbed by the economizers and there is a certain feed temperature above which it will not pay to install economizers to further heat the feed water. Air heaters for heating the air for combusion may be used to advantage in plants where the feed water is heated by abstracting steam from the turbine to a temperature that will handicap the economizers. The cost of fuel is a governing factor, and it pays to install a more expensive and complicated plant for a high-priced fuel and to shoulder such extra expense as may be required for operating the more complicated plant.

The great majority of the boilers built for higher pressures and temperatures have been for supplying steam at 350 lb. or thereabouts where the steam is not re-superheated between the stages of the turbine. Boilers having a total heating surface of about 2,560,000 sq.ft. were sold prior to 1924 for 54 stations for working pressures of 350 lb. or more. Of these, boilers having 400,000 sq.ft. were for 650 lb. working pressure. In the latter plants the steam is re-superheated.

The highest pressure for which the Babcock & Wilcox Co. is building boilers is somewhat over 1,200 lb. per sq.in., one such boiler being for the Edison Electric Illuminating Co. of Boston, and one for the Consolidated Safety Valve Co. The Commonwealth Edison Co. of Chicago, was the first to order a 1,200-lb. boiler, but the manufacture of it has been temporarily held up awaiting results of its 650-lb. development. The boiler for the Edison Electric Illuminating Co., of Boston, is arranged to evaporate about 150,000 lb. of steam per hour as a maximum. Few appreciate what it means to design and build such high pressure boilers. A small boiler can readily be constructed for operating at any desired working pressure, but it is a far different undertaking to design one to meet the exacting demands of central power-plant work.

In developing a boiler for 1,200 lb. working pressure for power-plant service, one company had before it the results secured in the tests of the series experimental boiler, as well as its experiments with the use of 1-in. tubes in a great number of marine boilers. It was finally decided to use tubes 2 in. in diameter, which is a convenient size for internal cleaning, and to employ a steam-and-water drum to provide a reserve water capacity, which is highly desirable in starting up a boiler

*Extract of paper read before The First World Power Conference, London, July 1, 1924.

†Advisory Engineer, The Babcock & Wilcox Co., New York City.

and in operating it under variable loads. The possibilities of operating without a steam-and-water drum were carefully considered. This would, of course, make it necessary to provide feed pumps having 100 per cent reliability. The relative merits and cost of the single steam-and-water drum for providing reserve water capacity as compared to a multiplicity of drums or tubes was also considered, and it was finally agreed that a boiler with 2-in. tubes and with a single steam-and-water drum was the best. This boiler will be described later on.

The limit of capacity ordinarily depends on the maximum amount of fuel that can be burned and the draft available for overcoming the resistance of the gases flowing through the boiler. It was considered good practice not very long ago to operate stationary boilers at capacities not exceeding an evaporation of about 5 lb. of water per square foot of heating surface. The maximum capacity called for has gradually crept up until today boilers are regularly installed to evaporate from 8 to 10 lb. of water per hour per square foot of heating surface and in some instances for a still higher rate of evaporation. These capacities are obtained with both coal and oil fuels and are sometimes considerably exceeded in actual operation.

Increase in fuel cost and a growing tendency toward base-load operation has led to a steady improvement in economy. It does not follow that a plant designed to give the highest thermal efficiency will have the highest commercial efficiency. Economizers will add to the thermal efficiency, and the same applies to air heaters, whereas on considering all features it may be that the character of the load, class of operation and fuel conditions, will make it inadvisable to install either economizers or air heaters.

Effect of Many Passes

Additional efficiency may be obtained by causing the gases to pass over a greater number of rows of tubes, and in some cases by changing the arrangement of the tubes. In the case of a boiler with horizontally extending tubes, adding to the height so as to increase the number of rows of tubes from, say, 14 to 20 will result in a considerable increase in efficiency with but little increase in draft loss. In certain instances the ability to operate the boilers with natural draft makes it desirable to use a high boiler without an economizer or air heater.

Increasing the number of passes increases the draft loss, much of the loss being due to the change in direction of the gases over the ends of the baffles. This has led to an endeavor to minimize the number of passes, and we have built a number of two-pass boilers. It may be asked, Why not use a single pass? The answer is that a single-pass boiler for a given capacity and efficiency would cost more than a two-pass boiler. The two-pass boilers also lend themselves more readily to the separation of soot and cinder from the gases, and fit in well with most plant layouts when used in connection with economizers and air heaters. A two-pass boiler costs more than a three-pass one for a given capacity and efficiency, and most boilers are three-pass. In the case of Stirling boilers the gases are made to pass three or four times lengthwise of the tubes, including the pass over the tubes next the furnace, and in some cases five times.

Wrought-steel economizers are used to a much greater extent than formerly, although subject to more interior corrosion than are cast-iron units. Interior corrosion of the steel economizer is now overcome by reducing the free oxygen in the feed water. At first it was thought that removing the free oxygen by heating to 212 deg. in an open heater would prevent this corrosion, but some corrosion was observed with the oxygen component as low as 0.2 c.c. per liter, and this led to installing an Elliott apparatus for removing the free gases from the feed water at its hotwell temperature. The water treated by this apparatus was found to contain practically no free oxygen, and there was no appreciable corrosion of the economizers after it was put in use.

In some small plants where steel-tube economizers are used, a considerable part of the free gases is eliminated from the feed water by heating it in an open heater to about the boiling point at atmospheric pressure before feeding it to the economizers. Corrosion in the economizers has been found when they are operated in this way, but with the counterflow type of economizer there has been little or no corrosion above the sixth and eighth rows from the bottom. This is, of course, an important advantage, as in case of corrosion it is necessary to replace only the lowermost tubes.

The economies now demanded have stimulated the use of air heaters. Most of the air heaters built by this company are made up of tubes because it has been found necessary in the case of plate heaters to weld all the joints to prevent leakage. For equally tight and efficient heaters the tubular heaters cost less than the plate heaters and are easier to repair.

Use of Air Heaters

The gain in efficiency due to the use of air heaters may be greater than that corresponding to the heat absorbed by the air heaters, on account of the effect of the heated air in increasing the heat absorbed by the boiler and improving the efficiency of combustion. In certain cases air heaters have the added advantage of increasing the available boiler capacity 10 to 15 per cent by making it possible to burn a greater amount of coal.

Furnace design must be co-ordinated with that of the boiler to secure the best results. From time to time methods have been suggested for measuring the efficiency of a boiler apart from that of the furnace, all of which have been approximate. The line of division between the stoker-and-furnace efficiency and the boiler efficiency is influenced by delayed or secondary combustion between the boiler tubes, which is an element that cannot be included in any analysis. The effect of excess air is another element that cannot be correctly divided between the boiler and furnace. The combined unit should, therefore, be considered in any study of efficiency or capacity.

With an incorrectly designed furnace it may be possible for a lane of gases containing excess air to pass through the furnace and boiler to the uptake. The effect of the eddying action within the furnace in mingling unconsumed combustible matter with the excess air must be carefully studied. An irregular mingling of the fuel and air may lead to pulsations within the setting.

Furnaces are made considerably larger than in former practice. For coal-fired boilers with forced-blast chain-grate stokers the latest installations have about 0.4 cu.ft. of furnace volume per square foot of boiler-heating surface. In the case of the latest powdered-fuel furnaces the volume is about 0.6 cu.ft. per square foot of

boiler-heating surface. The latest installations of blast-furnace gas furnaces have a volume of at least 0.2 cu.ft. per square foot of boiler-heating surface. These figures are for the larger furnaces used in stationary practice. With smaller furnaces a considerably higher capacity is often developed than these rules would call for; at peak loads 60 lb. of water or more may be evaporated per hour per cubic foot of furnace volume. In marine practice as high as 150 to 180 lb. of water may be evaporated per hour as a maximum per cubic foot of furnace volume in both coal- and oil-fired boilers. It is necessary to operate with the cleanest of feed water and with the highest class of attendance in order to secure such high capacities.

The question may be raised as to why furnaces are made, say, two or three times as large for stationary practice as for marine practice. The reason is that increasing the size of the furnace makes it possible to burn the fuel more completely within the furnace before the gases strike the boiler tubes. One cannot select fuels as carefully for stationary as for marine work, and where a fuel is of such a nature that it gives trouble through the accumulation of slag on the outside of the boiler tubes, the larger furnaces minimize the difficulty.

Relation of Efficiency to Furnace Temperature

A high efficiency of combustion involves a high furnace temperature, because the effect of increasing the amount of CO_2 in the gases leaving the furnace is to increase the furnace temperature. The brickwork employed today will fail in most cases if brought to the full heat of the furnace, and to maintain the brickwork there must be some cooling. In most instances enough heat may be dissipated through radiation from the sides of the furnaces combined with the radiation from the brickwork to the boiler tubes, without employing any special means for cooling the furnace walls. Where furnaces are operated under a suction, the cooler air from the outside percolates inward through the brickwork and serves to prevent overheating, whereas if operated under a pressure the hot gases percolate outward and increase the temperature of the brickwork. The difference in the temperature of the brickwork when running with a slight suction, say, 1 in. of water, and when running with a pressure within the furnace may cause all the difference between success and failure. The greater the load carried by the brickwork, the more likely it is to fail by plastic deformation. Firebricks of the best quality ordinarily used in boiler settings, which have a fusing point somewhat higher than 3,100 deg. F., may show plastic deformation under a load of 20 lb. per sq.in. at 2,000 to 2,200 deg. F. A reduction of the load to 10 lb. per sq.in. will increase the temperature at which the bricks begin to deform about 200 deg. F. As furnace temperatures of from 2,700 to 3,000 deg. F. exist with certain grades of fuel, it is apparent that care must be taken to cool the brickwork.

The larger furnaces in use today require special care in the design and the laying-up of the brickwork. The Babcock & Wilcox Co. usually builds the furnace walls entirely of firebrick, without red brick on the outside. A large wall tends to bulge inward toward the fire. Horizontal rows of bonding tile are held in place by cast-iron pieces which engage the sides of the tiles and also engage buckstays outside the walls in such a way that the wall can expand in any direction, whereas it is held in alignment and prevented from bulging either inward or outward. Another effective way is to incline the walls outward, making them rest against a steel framework.

In many instances air cooling of the walls is resorted to. Where the heated air is conveyed to the furnace, there is a double benefit, because in addition to assisting in the maintenance of the walls, the heated air improves the combustion.

In many cases boiler tubes are placed on the furnace or fire side of the walls for cooling the walls, these tubes being connected so that there is a circulation to and from the steam-and-water drum of the boiler. The use of furnace-wall cooling tubes has been growing, and they give advantageous results in protecting the brickwork where the cooling effect on the furnace does not interfere with the combustion.

In the designs so far tested, exposing more or less of the boiler surface to the direct radiant heat of the fire has a comparatively small influence on the efficiency.

The problem of employing furnace-wall cooling tubes is a highly complex one, and the best arrangement can be obtained only by painstaking investigations and tests. The way in which cooling tubes are connected to the boiler pressure parts must be carefully considered so as to provide proper circulation and avoid a water hammer on starting up, as well as to allow for the expansion of the tubes and the connections on being heated. Such cooling tubes should be used only with the cleanest of feed water, as they are usually the first to give trouble should a deposit collect on the inside.

Higher-class men are now employed in the boiler rooms of our large power plants. It is becoming appreciated more and more that, irrespective of how well a plant may be designed, the best results can be secured only when it is properly operated. There are still, however, glaring examples of inefficiency in this direction, and often the simplest requirements of good practice are disregarded. The trouble is seldom with the fireman, but is due to his lacking the proper means to guide him in his work. In some cases it would seem that the fireman is expected to work by intuition rather than in accordance with any method or system.

The fireman is human and cannot be blamed if he lacks interest when given a lot of arbitrary instructions which he is supposed to follow like an automaton without knowing the results. Where he can see the results of his work and knows that his efforts are appreciated, he will take an active interest.

In burning powdered coal, the handling of the ash is important. In some earlier installations there was trouble with ash and clinker fusing and solidifying at the bottom of the furnace into a mass which had to be broken out at intervals in a cold state, necessitating taking the boiler out of service. In the latest furnaces this is overcome by cooling the ash before it strikes the furnace floor to a point where it will not form a fuse mass.

The maximum temperature of superheated steam for operating steam turbines in power-plant practice is 700 to 750 deg. F. In either case some excess above the average operating temperature is permissible. Unless means are provided for regulating the superheat, there is bound to be some variation with a superheater placed within the setting, due to changes in the conditions of firing or irregularities in the feeding of the water to the boilers, aside from any variations that may come through a change in the capacity at which the boiler is operated.

Single-Pass Condensers Require Less Pumping Power Than Double-Pass

By PAUL BANCEL*

Neglecting external head, a single-pass condenser requires about one-eighth the pumping power for the same amount of cooling water as with a double-pass unit. Resistance of external piping and rearrangement of the condenser design, so as to utilize single flow to the best advantage, are here discussed.

SINGLE-PASS condensers are relatively old. In *Power* of Jan. 1, 1918, page 3, appears a photograph of a 70,000-sq.ft. single-pass condenser serving a 45,000-kw. turbine at the Detroit Edison Co. Also, in the Aug. 1, 1922, issue, is shown the single-pass condensers at the Gennevilliers station of Paris, serving 40,000-kw. turbines.

During the war a great many American destroyers were built with single-pass condensers having circulation forced by means of an external scoop. The same practice is followed in the English Navy, and excellent photographs of the single-pass condensers on the class "L" destroyer are included in the Aug. 1, 1919, issue of *Engineering*. The Nov. 14, 1919, issue also shows single-pass condensers for H. M. light cruiser "Dragon." The first condenser of a modified single- pass type built by the author was installed in 1917 and is mentioned farther on.

Consider the two-pass condenser as in diagram *A*, Fig. 1, using 1,000 gal. per min. and a friction head of 12 ft., or 6 ft. per pass. If this condenser is made single-pass it is evident that 2,000 gal. per min. can be pumped with 6 ft. friction, while the theoretical horsepower so far as the condenser is concerned remains the same. If reduced to the original 1,000 gal., the friction will then be one-fourth and the horsepower one-eighth.

In the two-pass, as at *A*, the horsepower can be represented by the equation, $hp. = K \times 1{,}000 \times 12$; and in the second case,

$$hp. = K \times 2{,}000 \times 6$$

By pumping more water through the same condenser, the capacity is naturally increased. Thus if the temperature rise with 1,000 gal. was 20 deg., it would be only 10 deg. with 2,000 gal. and the mean temperature of the water would be considerably lower, so that the capacity of the condenser might be increased 20 or 25 per cent.

These are the advantages of single-passing under ideal conditions. Practically, several factors enter into the problem, calling for special features of design if a single-pass condenser is to be highly efficient.

In the first place no condenser installation can be made without a frictional loss external to the condenser. This must be added to the internal condenser friction. Therefore, the horsepower must depend on the total head, and if we assume an external friction and static head of 8 ft., the following obtains:

Condenser *A*, two-pass,

$$hp. = K \times 1{,}000 \times (12 + 8)$$

Condenser *B*, single-pass,

$$hp. = K \times 2{,}000 \times (6 + 8)$$

The foregoing gives a ratio of horsepowers of 20 to 28. It is evident, therefore, that for the same horsepower somewhat less than 2,000 gal. can be pumped in condenser *B*.

Secondly, the additional water quantity that can be pumped for the same horsepower will only give the decrease in mean water temperature and increase in condenser capacity as calculated, provided the condenser design is such as to utilize the water. It is well known

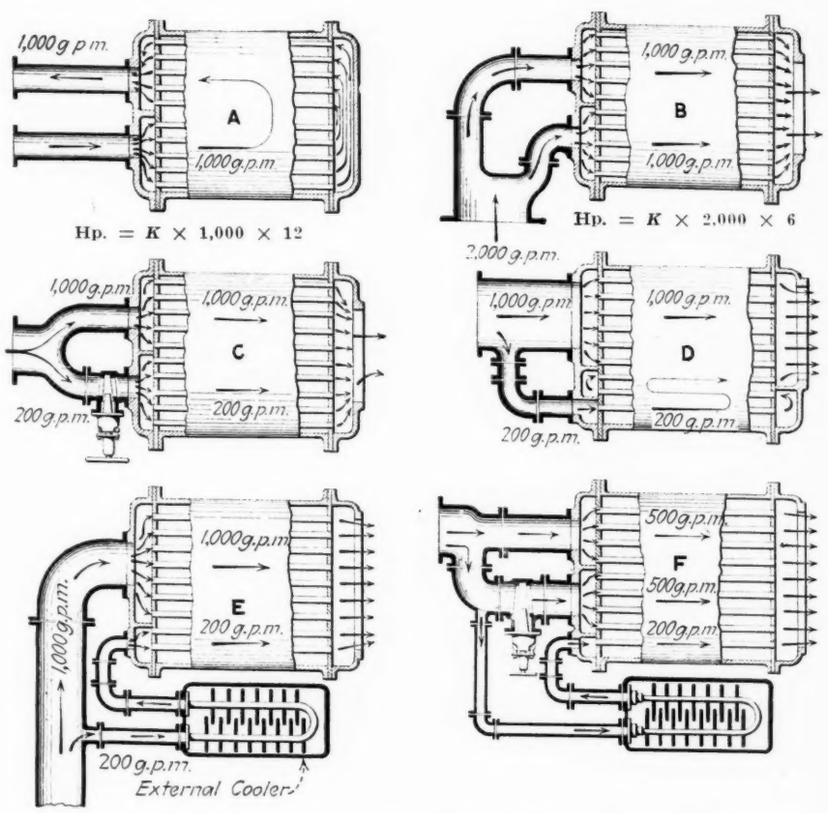


Fig. 1—Cooling water paths of single- and double-flow condensers

Diagram *A*, is a double flow which receives one-half the water as the single flow *B*, with equal internal pumping power. Water is more efficiently used in *C* than in *B*, and to still better advantage in *D*. External coolers at *E* and *F* improve single-flow condenser tube efficiency.

*Condenser Department, Ingersoll-Rand Company.

that in a condenser some of the tubes nearest the steam inlet do a great deal more work than others in the bottom and near the air outlet.

A reproduction of one of the condensers on the U. S. battleship "Maryland" as reported in the November, 1922 issue of the A.S.N.E. Journal is shown in Fig. 2. Two tests may be selected as in the table.

FROM TESTS OF BATTLESHIP CONDENSER

Square feet cooling surface	12,050	12,050
Horsepower	10,949	16,403
Steam load, lb. per hr.	108,320	166,149
Steam temperature, deg. F.	81	86
Injection temperature	47.9	47.1
Temperature rise (first pass)	0	0.6
Temperature rise (second pass)	11.9	16.3
Amount of cooling water, gal. per min.	18,500	19,200

The temperature rise of the water in the bottom of the condenser is negligible, and all the temperature rise occurs in the top. Now, if this condenser were arranged for single pass, it would take twice the water quantity to do only the same amount of work. There would be no increase in the capacity because all the water going through the bottom pass would not be heated. It would require 19,200 gal. in the top pass, which would heat 16.3 deg. and there would be another 19,200 gal. going through the lower pass, not heating at all.

The pump with 38,400 gal. per min. would require extra horsepower, however, because, as pointed out, the external head to the condenser cannot be neglected. Assuming that the head relations are such that the same horsepower would be used when pumping 170 per cent water instead of twice the water, there would then be 85 per cent of 19,200 gal., which equals 16,300, going through the top pass and another 16,300 gal. through the bottom pass.

The reduction in water quantity in the lower pass would not cause any loss because these tubes are not doing any appreciable work. In the upper pass, however, there would be only 85 per cent of the water quantity going through each tube, and this water would actually have to be heated further, which would raise the steam temperature. Furthermore, the heat transmission in the tubes would be poorer because of the lower velocity, and for these two reasons there would be a considerable reduction in vacuum. This condenser would, therefore, give very much poorer capacity or vacuum if it were made single-pass.

In the same way, if the condenser were supplied with only the same total quantity of water, the water in the top would now be only 9,600 gal. and the condenser would be greatly reduced in capacity and the loss would be very much more than the saving in power for pumping the water.

It is manifest now that in such a condenser it is necessary to keep up the water supply to the tubes in the top which the steam reaches and that in the bottom tubes the water supply can be choked. Thus in diagram *C* a throttle valve can be used to cut down the water to the lower half of the condenser from 1,000 gal. to, say, 200 gal. With a condenser of the type on the "Maryland," decreasing the bottom water supply would not appreciably affect the capacity. Taking the simple conditions as in diagram *C*, the horsepower would now be $K \times 1{,}200 \times (6 + 8)$ and the ratio of horsepower as compared to diagram *A*, Fig. 1, would be as 16.8 divided by 20. The same work would, therefore, be done with less horsepower.

It is also evident that a similar result can be obtained by various other devices; for example, putting small orifices at the entrance to the tubes of the lower pass or again, using a smaller-diameter tube in the bottom pass of the condenser.

The throttle valve in diagram *C* represents considerable waste of power which could otherwise be used to advantage in giving better distribution of water to the tubes. Thus in *D* the lower section is divided into three passes and the water quantity automatically choked to 200 gal. by having to make three turns through the condenser.

Some years ago one of this type was built by the author. The upper pass contained 40 per cent of the tubes and the three lower passes each 20 per cent. The two lowermost passes, comprising 40 per cent of the tubes, showed hardly any rise in water temperature. The third lower pass showed a fair rise. The top tubes passed almost all the water and did about 80 per cent of the work on 40 per cent of the surface.

In the present type of single-pass condenser, which is an evolution of that mentioned before, the two lower-

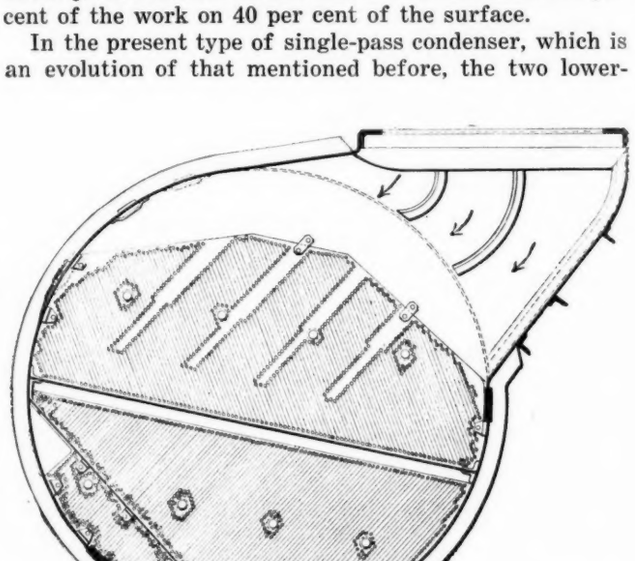


Fig. 2—Double-pass condenser where little or no work is done in the lower pass

most passes of diagram *D* are replaced by external grid air coolers as described in the May 13 issue.

The water circuit is therefore, as shown in diagram *E*, Fig. 1. There is a single pass through the top and then at the bottom a two-pass circuit made up of the external cooler for the first pass and the lowermost tubes for the second pass.

The evolution of the condenser as herein described has been to concentrate the water where it will do the most good. The zone where this greatest activity occurs is not fixed. In other words, the distance into the condenser that the steam penetrates varies with conditions. In winter time it is less than in summer. This leads to the final arrangement as in diagram *F*, where the top section is split in half and 500 gal. go to each part separately. A throttle valve is shown for the lowermost of the top sections so that under winter conditions the 500 gal. can be reduced independently of the flow in the top. In actual practice this arrangement works to advantage chiefly with large condensers where two pumps are used, and the regulation of the water is obtained with the separate pumps supplying the separate sections.

Flow Meter Needed for Refrigerating Plant

THE refrigerating engineer is at all times working more or less in the dark as to the efficiency he obtains in the plant. While meters for measuring the flow of steam have been improved to a point where the errors of such instruments are very small, no servicable meter has been devised to measure the flow of ammonia. Some attempts have been made with venturi meters and weighing machines, but the latter are impossible for daily records, while the former has been pronounced inaccurate when the flow is variable.

The instrument most needed is one that can be placed in the suction or discharge of an ammonia compressor to record the gas flow per minute. With modern equipment whereby all liquid ammonia is trapped off before reaching the compressor, the fluid to be measused would be a vapor or gas. Variation in suction pressures and temperatures necessarily would need be compensated for by some device, but this does not seem impossible.

The objection to the measurement of the liquid ammonia leaving the condenser is based on the tendency for the liquid to come down in slugs similarly to the water of condensation in many heating systems. This is indicative of poorly designed condenser apparatus, and the trouble does not or should not exist in shell and tube condensers.

In the face of the absence of instruments capable of measuring the work done in a plant, it is surprising how near to the maximum possible results many plants go in every-day operation.

The Water-Wall Furnace

EXCEPT for brief news references, the article "Water-Cooled Furnaces Make Records at Hell Gate," on page 354, is the first published description of the water-wall furnaces installed and tested several months ago at Hell Gate. The previous reports that an over-all efficiency of 92.7 per cent had been obtained and that the boiler had been forced up to 528 per cent of rating for a short period (since raised to 603 per cent) led to much speculation as to the details of these furnaces and the reliability of the tests.

As far as can be judged from the report, the tests were made with considerable care and under competent guidance. In the attempt to locate the cause or causes of the remarkable efficiency, the heat balances shown in Table II of the article are helpful.

In Test 5, where the highest efficiency was obtained, 84.6 per cent of the heat in the coal was absorbed by water and steam in the boiler and superheater. Another 8.1 per cent was absorbed in the economizer, leaving 7.3 per cent as the total of the losses. Of this 3.1 per cent was due to the moisture from the combustion of the hydrogen in the fuel (an unavoidable loss), 3.2 per cent to the sensible heat of the dry gas and 1 per cent to all other losses combined. A total of 4.2 per cent for controllable losses is remarkable. From the test figures it appears that loss of combustible was almost completely eliminated. The extraordinarily low dry-gas loss was obtained not so much by cutting down excess air (about fifty per cent was present at the economizer exit) as by lowering the flue temperature to 193 degrees F. In this of course the large economizer was an important factor, but even without the economizer the efficiency was 84.6, an extremely high figure for a stoker-fired boiler. The dry gases entering the economizer contained 9.8 per cent of the heat of the fuel. The cooling of the gases in the boiler proper to within 74 degrees of the saturated-steam temperature indicates excellent heat absorption, for which the water-cooled walls are presumably responsible.

This is particularly interesting in view of the fact that the original reason for installing walls of water tubes was to reduce furnace maintenance rather than to increase efficiency. In fact the installation was made in the face of predictions that good combustion would be impossible in a furnace whose walls were composed so largely of relatively cold surfaces. In view of the performance of this furnace it is profitable to speculate on the feasibility of still further reduction of the refractory areas. Possibly there is a limit beyond which this cannot be carried without loss of efficiency, but this limit has not yet been determined.

These tests show the importance of re-examining old ideas to see if they are really sound. The case of superheated steam in closed heaters is similar. It has always been held that superheated steam reduced the capacity of heaters, but actual tests failed to prove it. As a result engineers have been forced to revise their opinions in this respect.

When in Doubt Adopt the Safest Course

NO INDUSTRY is more progressive than that which involves the electrical generation and transmission of power, and as a result every new project generally involves new and untried features. In the steam plant during the last two or three years a number of different systems of burning powdered coal have come into use, steam pressures up to twelve hundred pounds are being adopted with exceedingly high temperatures and reheating between the high-pressure and low-pressure turbines. The adoption of the regenerative cycle has also helped to complicate the problem, as well as the adoption of larger units than had been used before. All these and many other innovations in steam power-plant design carry with them problems that, although they may not be totally untried, have afforded little opportunity to gain from experience. In putting into effect these improvements, the designers of equip-

ment and plants must endeavor to supply in sound judgment what is lacking in practical experience.

This, which might be called evolution, has been going on in the steam power plant and has also been taking place in hydro-electric plants. New designs of runners have been developed, new types of draft tubes have been devised and applied to the largest units. Units of sizes not dreamed of a few years ago have been built and put into successful operation. The world record capacity of Francis wheels has been doubled for the same head within the past four years. The so-called propeller-type turbines, which less than ten years ago were built in sizes of one hundred horsepower, have reached capacities of over thirty thousand. The use of higher heads and larger volumes of water has brought new problems in pipe-line, penstock, valve and surge-tank design and construction, on most of which there has been little practical experience to guide the designer. Here, as in steam plants, what has been lacking in practice has had to be supplied by judgment.

For the most part these problems have been successfully met. Where failures have occurred, they have generally been of a nature that could have been avoided if a more conservative design had been used. A case in point is the cracking of a concrete-lined surge tank. This might have been prevented at little added cost by properly reinforcing the concrete during construction. Other failures in concrete structures have occurred that could have been prevented by the expenditure of a few hundred dollars for additional reinforcing when the work was first done. Failures of penstocks and other parts of the equipment have occurred that might have been prevented by a comparatively small expenditure when the plant was built and the expense and inconvenience incident to the failure avoided.

Probably because of the general belief that water power should be cheap power, it frequently happens that there is a tendency to cheapen the construction of such installations, even to an extent that interferes with their reliability and efficiency. Such practices cannot be justified on economical grounds, since the increased maintenance costs and the loss of revenue incident to having a machine out of service may soon involve a greater expense than that required to have made the construction safe for the worst conceivable conditions. Some recent experiences have proved the soundness of such a procedure.

Blocking the Emergency Governor out of Service

WITH the large number of steam turbines now doing duty it is perhaps not strange that some operators should run the risk of a possible disaster by deliberately rendering inoperative the emergency-valve closing devices. The fact that a small turbine-driven pump recently became wrecked from such a cause is evidence that this undesirable practice exists to some extent. Because of the unfortunate result in this instance, that the engineer paid the forfeit with his own life, some may be deterred from deliberately taking chances of this sort.

On the other hand, there may be unusual conditions such as that of an overspeed protective device being unavailable owing to lack of necessary parts, etc., and yet it is deemed necessary to operate by those in general charge and qualified to judge. Sometimes conditions arise that cause slight overspeeding at frequent intervals. Delays incident to closing the throttle, setting the emergency valve and starting up again become so harassing that the alternative of eliminating the overspeed governor may be seized upon against better judgment so as to avoid frequent shutdowns.

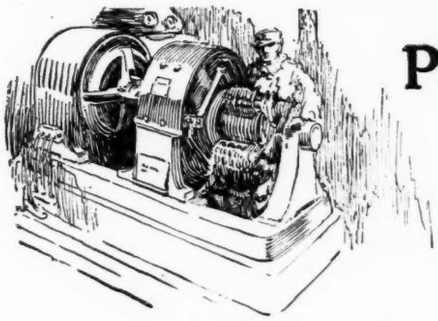
The lesson from several such accidents indicates that much trouble could have been prevented if the operator had kept in mind the idea of avoiding the risk of operation without additional overspeed protection, even if the unqualified rule against blocking the emergency were disregarded. For instance, serious overspeed has hardly ever been known to result if a man was stationed at the unit prepared to make a quick shutdown in the event of trouble. Not infrequently, auxiliary devices acting in the capacity of overspeed governors may be constructed for temporary service. While in rare instances there may be valid reasons for operating without the prescribed emergency device, a self-respecting engineer could not fail to provide some other reliable means in its place for checking overspeed in a steam turbine.

The first duty of an operator is that of maintaining both the main governing mechanism and the device for protecting against overspeed in a normal condition. If overspeeding in operation results from other causes, such as the loss of suction in a turbine-driven centrifugal pump, or lower-pressure steam entering a turbine through drains or extraction lines, needless to say the importance of remedying it in the best way and at once is scarcely less important. Where it is necessary to contend with unusual circumstances, efforts should be made to solve the problems in a safe way, as "taking a chance" is considerably less a matter to be proud of with modern highly organized equipment than with that of an older period. In fact, ingenuity applied to safeguard operation frequently means more and receives greater attention than that expended in other channels.

The announcement of the Franklin Institute's Centenary Celebration brings to mind the important service that this institution rendered in its early investigations to learn the causes of boiler explosions and to suggest reforms to guard against them. The first numbers of its official organ, the *Franklin Institute Journal*, were published during the time, 1826-7, when boiler explosions were prevalent on the newly invented steam boats which resulted in much loss of life. The report of the institute's committee which investigated these explosions caused several essential reforms to be inaugurated.

Under the title, "Politics and Coal," the Mining Association of Great Britian makes a spirited attack upon the Lloyd George coal report. Evidently, the British are about to be treated to a smoke screen of charges and countercharges calculated eventually to block any definite solution of their coal problem. If this be true, they may be assured of a sympathic understanding from the American public.

Recent casualties resulting from the presence of carbon monoxide in powdered-coal bins brings up a new hazard that will have to be reckoned with in the design and operation of such installations. Fortunately, means for combating the danger are now available, and it remains only for strict insistence on their enforcement.



Practical Ideas *from* Practical Men

Rules for the Construction of Unfired Pressure Vessels in Holland

Unfired pressure vessels, like steam kettles and evaporators, are not subjected to any inspection if, in case a liquid is heated, the contents do not exceed a volume equal to 21 cu.ft. and the absolute steam pressure is equal to or less than 7.35 lb. per sq.in.; and in case the compartment wherein the liquid is heated is separated from the steam space, the steam pressure cannot exceed 7.35 lb. per square inch.

If no liquid is to be heated, the contents should not exceed 35.3 cu.ft. and the absolute steam pressure 14.7 lb. per square inch.

Unfired pressure vessels should be fitted with:

1. One safety valve when the pressure is just negative to the boiler pressure. However, if the vessel pressure is less than half the maximum boiler pressure or the boiler steam is in direct contact with the liquid, two safety valves are necessary.
2. A small test cock, in order to make sure whether the pressure is on or off. It should be possible to prevent this cock from being clogged while the regular steam pressure is on.
3. A steam gage.
4. Manholes and handholes in order to facilitate inspection.

The evaporators are to be provided with:

1. Two safety valves, computed according to the formula,

$$M = 36\sqrt{\frac{W}{d + 0.62}}$$

in which M is the diameter of valve in millimeters, W the heated surface in square meters, and d the maximum absolute steam pressure in kilograms per sq.cm.; the ultimate value of d taken equal to 1 (one kg. per sq.cm.). The diameter of the valve should be at least 2 in.

2. A steam gage.
3. A water gage.
4. A blowoff cock.

The diameter of safety valves fitted to unfired pressure vessels, exclusive of evaporators, is sufficient when the relieving area of the valve exceeds the relieving area of the steam pipe connecting the vessel with the steam boiler, 25 per cent. The diameter should be at least $1\frac{1}{4}$ in. but need not exceed $2\frac{5}{8}$.

The safety valves and the steam gage should be placed directly on the pressure vessel or on the steam branch connecting the steam main with the pressure vessel.

If a number of unfired pressure vessels are connected to a common steam branch, it is sufficient to place one safety valve and one steam gage on that steam branch unless in one or more of these vessels a liquid is heated while being in direct contact with the steam, or the maximum steam pressure in one or more of the vessels is less than 50 per cent of the maximum boiler pressure. In these cases two safety valves are required.

When a liquid is heated in a separate compartment, a safety valve is required if the vapor pressure may exceed 7.35 lb. per square inch.

If necessary, this compartment is to be provided with a test cock, a steam gage, manholes and handholes. The diameter of the safety valve is found by the formula.

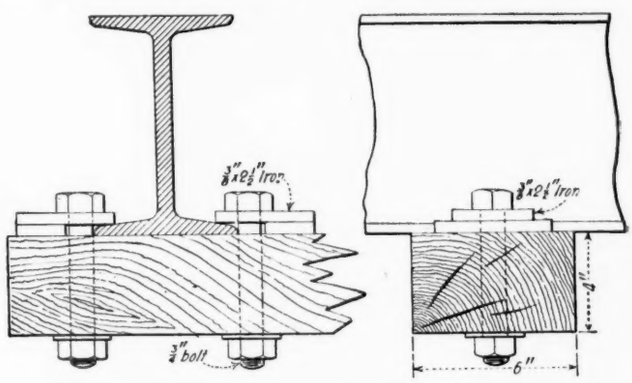
$$M = 13\sqrt{\frac{W}{d + 0.62}}$$

where M is the diameter in millimeters, W the heated surface in square meters, and d the absolute vapor pressure. The diameter of the valve seat should be at least $1\frac{1}{4}$ in., but need not exceed $2\frac{5}{8}$ in. The inlet opening of the valve should be protected in order to prevent clogging up.

Schoonhoven, Holland. H. WIELAND LOS.

Supporting Motor from Ceiling I-Beams

The method of supporting a motor from a ceiling will depend somewhat upon the conditions encountered. In the figure is shown a way that was used to attach



Method of attaching wooden beams to steel I-beams

wooden beams to steel I-beams for supporting a motor.

If facilities are available, various kinds of clamps can be built for making the connection to the I-beams, but in this case the job had to be done in the simplest manner possible and with the material at hand, which happened to be $\frac{3}{8}$x$2\frac{1}{2}$-in. iron. This was cut into pieces about 8 in. long and $\frac{7}{8}$-in. holes bored near the middle for $\frac{3}{4}$-in. bolts. The holes through the wooden beam were bored so that the bolts fitted close to the I-beam; thus the only function of the iron strips is to sustain the vertical load. CHARLES A. ARMSTRONG.

New York City.

Avoiding Water Pockets in Steam Lines by the Use of Eccentric Reducing Fittings

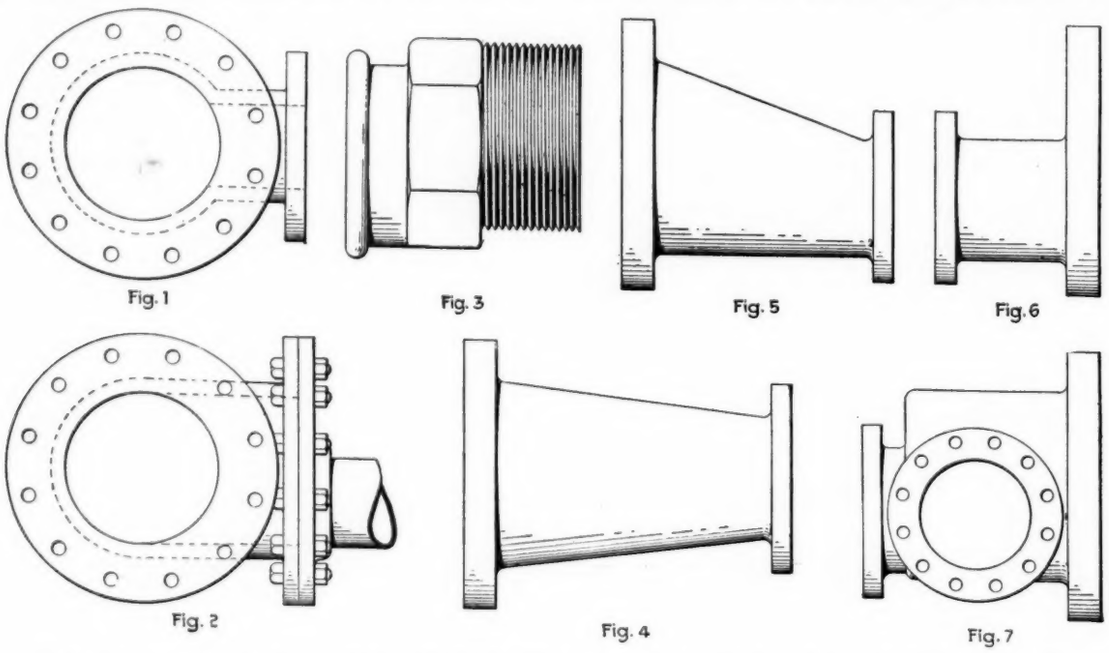
Every effort should be made, when installing either large or small pipe lines, to avoid the possibility of pockets appearing in which water may collect, as many costly accidents have happened on account of water being picked up by a sudden increase in the amount of steam required to do the work, thus causing slugs of water to be carried over with the steam. One common cause for these defects is found where pipe sizes are reduced, either on the line or where branches are needed.

A sample of this lack of foresight is found in my engine room, where two branch lines are taken to supply high-speed direct-connected units. In both cases only ordinary flanged tees were used that are full size on the run and reducing on the outlet, as shown in Fig. 1. So far as the first engine is concerned, this is a good feature, because any water that may result from condensation, flows along the bottom of the pipe and cannot get into the outlet, but when it reaches the second tee, which is fitted with a blank flange on the run, the water can collect, as no outlet is provided for it unless a sudden rush of steam carries it over to the engine, which is liable to cause a wreck. A separator is in the branch line that disposes of this water, but every precaution should be taken to guard against these accidents, which are far too common, even with all the appliances on the market at the present time.

A tee with the outlet full size might have been used, with the eccentric reducing flange shown in Fig. 2, giving complete and automatic drainage. This flange may be used as part of a regular flange union if required on a straight line or wherever it may prove desirable. If screwed fittings are used instead of the flanges illustrated, the same effect is secured by the use of the ordinary eccentric-screwed bushing or offset-reducing coupling illustrated in Fig. 3. The male end is full size, but the female end is reduced according to requirements and is in eccentric form to provide perfect drainage at all times.

A taper reducer may be used on a vertical line, or even on one that is inclined at an angle of 45 deg. more or less, but it should be used only on a horizontal line when steam is to pass from the smaller to the larger end. If the direction of flow is reversed, a pocket will be found in which water will accumulate. Fig. 4 illustrates a flanged taper reducer. The fitting shown in Fig. 5 is called a flanged eccentric taper reducer and is not subject to the objection made to the preceding illustration, as it may be used in a horizontal pipe line with equally good effect provided it is located as shown. The lower side is at right angles to the face of the flange. If it was given one-half turn, it would not allow water to drain out continually.

This fitting gives a long gradual reduction of the cross-section of the pipe line, hence but little friction results, but there may not be enough space available to install it, therefore it may be necessary to use the short fitting shown in Fig. 6.



Figs. 1 to 7—Types of eccentric reducing fittings used to prevent water pockets in steam lines

Fig. 7 illustrates an eccentric reducing tee that is an excellent fitting for use where the line is to be reduced, also the outlet. Frequently, the line is run full size for the entire length and branches connected where they are wanted. Suppose that such a line is to supply steam for two turbines or engines. It might be carried full size to the first outlet, then this reducing tee should be used, the branch going to one engine and the run carried to the second. This not only saves on first cost, but also reduces the radiation losses.

I am well aware that these special fittings may not always be available at short notice, hence a pipe line may be installed with whatever can be secured, so long as it will convey the necessary amount of steam, but by making calculations in advance and ordering the materials a reasonable time before they are actually wanted for use, they can be secured under normal conditions of manufacture and delivery.

W. H. WAKEMAN.

New Haven, Conn.

Comments from Readers

Cause for Pressure Falling When Feeding Water to the Boiler

In the Aug. 12 issue Frederick Pakes tells of a case where the steam pressure would fall excessively when water was being fed to the boilers, and what he did to provide a remedy. If he had substituted a constant feed for the intermittent feed, the remedy would have proved satisfactory, without further trouble. If the change is made now, it will be a great improvement.

I know of no better authority on these matters than the Hartford Steam-Boiler Inspection & Insurance Co., which has published an article in which the spray feed is recommended. A short extract from this article follows:

> The present article is designed to show how all expansion and contraction of shell plates may be avoided and deposit of scale largely prevented. This can be accomplished in the writer's experience by spraying the feed water into the steam space of the boiler. Experience has shown that this can be done without noticeably increasing the moisture in the steam passing over to the engine.

This provides an answer to the question that he asks in conclusion.

W. H. WAKEMAN.

New Haven, Conn.

Electric Boiler Explosion

I read with interest the editorial in the July 22 issue concerning a recent explosion of an electric boiler. It is to be hoped that the results of a thorough investigation into the causes of the explosion will be given full publicity.

It has long been established, although it does not appear to be generally known, that decomposition of water takes place not only with direct current, but also with alternating current. While in the case of direct current the rate at which oxygen-hydrogen is generated is determined, after Faraday's "First Law," solely by the intensity of the electric current passing, in the case of alternating current the rate is smaller than would follow from Faraday's law, and is, moreover, dependent upon the frequency of the current, its density at the immersed surface of the electrode and the nature of both electrodes and electrolyte.

That the decomposition is smaller with alternating than with direct current is easily understood if one considers that, in consequence of the continual reversion of direction of current, hydrogen and oxygen are both evolved in rapid succession in one and the same place and that hence the conditions are favorable for the re-formation of water out of the components, which are in the "nascent" state. Tests have shown that in certain cases the rate of decomposition with alternating-current of commercial frequencies amounted to 30 to 50 per cent of the one calculated from Faraday's law, or in other words, that would take place with direct current.

In an electric boiler the oxygen-hydrogen gas thus liberated mixes with the steam generated. If the mixture were consumed uniformly as it is generated, the gases could never, in a commercial boiler, attain the concentration that is necessary to render the mixture explosive. In the ordinary case, however, steam condenses in certain parts of the system, while oxygen-hydrogen gas, possibly mixed with a small amount of air, is left as a dangerous remnant. In the instance reported it seems that the steam condenses right in the boiler, serving to heat the water. In this circumstance an explosive mixture is bound to accumulate above the water level. It stands to reason that here the possibility of accidental ignition, and hence of explosion, is far from being remote.

All things considered, the explosion theory, which has apparently been advanced, should not be lightly dismissed. There has been too much groping in the dark in this field, and it is not unlikely that the small number of accidents hitherto recorded may be ascribed more to chance than to real absence of impending danger.

ARTHUR A. BOELSTERLI.

Tompkinsville, Staten Island, N. Y.

Unaflow Engines Were Built in the Year 1827

In *Power*, May 6, 1924, appeared a brief history of the Perkins family, in which, unintentionally, the statements discussing the unaflow engine were so couched as to convey the impression that Jacob Perkins merely patented but never built such an engine. On the contrary, the periodical press of 1827-28 abounds in information proving that Perkins built his first single-expansion unaflow in 1827 at the engine works of Perkins & Co., Water Lane, London, adjoining the present site of the *Times* printing offices and quite close to the Black Friars bridge. Unaflow locomotives were built in 1843 according to a press paragraph dated April 18, 1827. The unaflow stationary engine was visible for public inspection every Saturday, and various views of this engine were published during that year and later.

This engine or one similar to it was moved in 1827 to the St. Catherine's Docks where it was used to pump out the excavations. It was erected alongside two Boulton & Watt engines, one of 16 and one of 10 horsepower, and was connected to the same pump shafting, it being rated at 30 horsepower. Its hourly coal consumption was, by actual weighing, certified to be 45 lb., or, assuming full load conditions, 1.5 lb. per brake horsepower. The running cutoff was 12 per cent, but a simple screw permitted the cam to trip the tappet down to 6.5 per cent for low powers. The steam pressures ranged from 200 to 300 lb. per sq.in. on this slow-running pump work; the usual pressure was 500 pounds.

The first compound unaflow was constructed in 1827 at the Perkins new engineering works, Regents Square, Kings Cross; the building is still standing with the firm's name showing along the front This compound unit, of which many descriptions were published in 1827 was stated to be a very considerable improvement over the Perkins single-expansion unaflow; steam-extraction was not employed as there was a separate

heating plant. Still another type was a twin-cylinder single-expansion marine unaflow arranged for direct connection to the paddle-wheel shaft.

Perkins established high claims for the compound unaflow engine, although he advocated unaflow for both simple and compound operation. Incidentally, both on European and American railways, the modern compound locomotive has shown marked superiority over the single-expansion engine (12-13 lb. steam per indicated-horsepower hour as against 18-20 lb.), and although abandoned for divers reasons, the drift in locomotive practice since 1913 is once more toward compounding, the three-cylinder design being preferred in German and English speaking countries until inadequacy of cylinder volumes compels return to the four-cylinder types. So advantageous has compounding proved, that British builders of biflow compounds at Wembley guarantee a lower steam consumption than do the makers of the single-cylinder unaflow. To combine the efficiencies obtainable by compounding and by unaflow construction, many British engine builders have, since 1920, reintroduced the Perkins compound unaflow system under the name of "heat-extraction engine," and two or three different makes are shown at the British Empire Exposition; Musgraves exhibit a specimen wherein steam may be bled from the receiver for industrial processes whenever necessary. A novel regulator has been introduced by which, whenever demand on the receiver for process steam exceeds a predetermined amount, that demand is automatically disallowed until the governor lengthens the high-pressure cutoff. In this way the balance of the mean effective total load in the two cylinders is not disturbed. Obviously, biflow compounds permit the same economy by steam tapping of the receiver as do unaflow compounds.

The question arises, How, then, was it that engines claimed to be so economical were abandoned? Only a knowledge of the actual conditions of the steam-engine market at that period permits one to understand why the best of all engines were then rejected by users. The general deification of Watt then prevailing, and the devotion of the contemporary writers to the Boulton-Watt interests, assured the continuance of the Watt engine until, year by year, the theories of Doctor Robison and others that the "same power could be obtained from one cylinder that could be extracted from two with the same steam operating in stages" was demonstrated by practice to be a complete fallacy because the *negative* horsepower indicated in single expansion had not been understood. By 1830 the unfortunate Hornblower had been fully vindicated; and later the wasteful operation of single-expansion was shown by McNaught, who "Mcnaughted" compound Watt engines in reducing their consumption by nearly one-half.

But Perkins came forward with his cylinder which formed its own exhaust valve by the piston uncovering the exhaust ports; and the only survivals of large-power engines are those employing the Perkins unaflow system. Perkins first unaflowed Watt cylinders with wonderful economy, and he unaflowed Hornblower compound cylinders with still more remarkable economy. The technical press of his day, in certain places, accused him of merely imitating those two systems with his own modification appended.

Since 1828 no such simple applications of the unaflow engine have been made. On the contrary, present-day makers and patentees follow the tandem unaflow type of cylinder, first employed about 1843 by G. & J. Rennie, of Blackfriars, London, and which the writer thinks was excusable as an expedient in those far-off days for a locomotive, but indefensible in stationary engines on the ground of its spread-out arrangement dissipating heat where it is wanted and conserving heat where heat-irradiation is sought to save water in the condenser.

No page in the history of the steam engine nearly approaches in interest the experiments of Perkins with unaflow engines working up to 1,600 lb. and 2,000 lb. pressure. This page in history is totally unknown to engineers at the present day and in view of its possible future service to the trade the writer hopes to be able to give transcription of the ancient documents treating of this matter.

London, England. CHARLES R. KING.

Large Saving Possible in Industrial Plants by Use of High Steam Pressures

The fact that steam pressures up to 600 lb. are economical and profitable has been proved by large central stations, but to date we have not heard of any such plants being installed for industrial purposes. But an industrial plant, whose demand for process steam exists twenty-four hours per day, as in paper mills, will show a larger percentage of gain than public-utility plants. I recently visited a paper mill that had just installed new equipment in their old power plant and they consider that they have the most efficient type of plant. It consists of three 3,500-sq.ft. water-tube boilers with no superheater, a 1,000-kw. turbine and four small turbines of 85 hp. each driving four paper machines and exhausting at 13 lb. back pressure into the driers. The boiler load is 30,000 lb. of steam per hour, of which 13,000 lb. is used by the small turbines, which carry a load of 65 hp. each and use 50 lb. of steam per horsepower-hour. The main unit has a water rate of 17.5 lb. of steam per kilowatt-hour at full load with 190 lb. steam pressure and 28 in. vacuum. The load on this unit is about 750 kw. and at three-quarter load the steam consumption is 14,500 lb. per hour, and the auxiliaries, consisting of a turbine-driven feed pump and turbine-driven air and circulating pumps, use 2,500 lb. per hour.

A study of the conditions in this paper mill discloses two significant facts which show clearly the direction in which the best economy lies. The process steam amounts to 43.33 per cent of the total load on the boilers, and the greatest loss in this plant is the condenser loss. To eliminate this loss we can adopt a method that any industrial plant of this class will find profitable. We can raise our steam pressure and superheat high enough so that 13,000 lb. of steam per hour passed through a non-condensing engine will generate our total power requirements. This demand for 13,000 lb. of steam per hour is the minimum steam consumption with any system of power generation, and the demand for this steam would exist even if all power was purchased.

Now the total power requirements with a high-pressure plant will be as follows: 750 kw., which represents the mill load, 260 hp. for paper machines and 45 hp. for auxiliaries. For best economy the auxiliaries and paper machine will be motor-driven and all the power will be generated by the main unit. This will add about 230 kw. to the load on the main unit, making a total load of 980 kw. To generate 980 kw. with

13,000 lb. of steam per hour, we must secure a water rate of 13.26 lb. per kw.-hr., and to get this low water rate with 13 lb. back pressure we must go to around 600 lb. steam pressure and considerable superheat, with possible reheating between cylinders. As a low water rate is necessary and it is also necessary to operate non-condensing, a reciprocating engine will be the logical choice, with a specially designed high-pressure cylinder expanding steam from an initial pressure of 600 down to around 150 lb. gage and then a low-pressure cylinder of standard unaflow design utilizing the steam at 150 lb. and exhausting against 13 lb. back pressure.

The cost of such engines and boilers will be high, but in view of the fact that the reduction in steam consumption is $\frac{30,000 - 13,000}{30,000}$, or 56.66 per cent, any plant, (where conditions are similar to the one herein described) can afford to pay three times as much per unit of capacity for the 600-lb. boiler as for 200-lb. ones. The cost of power and process steam in the first plant is as follows: Coal at $7.50 per long ton, 35 tons per day 302 days per year, $79,275; interest, insurance and taxes, maintenance and depreciation, 16 per cent on $92,000 (present plant cost $92,000 two years ago), $14,720; boiler-room labor (6 firemen, 3 coal and ash handlers), $14,000 per year; total cost, $107,995 per year.

To arrive at the power cost in a 600-lb. plant, we must make an estimate of the capital charges, which will be greater, the fuel cost, which will be less, and the boiler-room labor, which will be less than in the first plant. Starting with the investment required for the 600-lb. plant, the boiler cost could be kept down to what the 190-lb. pressure boilers cost, since the load is only 43.33 per cent as much in the proposed plant and small boilers could be installed and operated at a high rating. To generate 13,000 lb. of steam per hour, two 2,000-sq.ft. boilers should be installed, and at 200 per cent rating the load could be carried on one of them. Only the pressure parts of the boilers and the superheater would be expensive, the settings would be much cheaper than in the 190 lb. pressure plant on account of the greatly reduced size of units, and the same with the economizer, which could be low pressure. The piping would be greatly reduced in size, and only one short high-pressure line from boilers to engine and the feed line would have to carry 600 lb. pressure. The present boiler plant, with feed pump and heater, cost $48,000, and the cost of the 600-lb. plant could be kept down to this amount.

In the engine room the present equipment cost $44,000 with piping. The new engines would cost more, but the piping would cost less (as there will be no steam-driven auxiliaries and no high-pressure lines to paper machines). Also we save the cost of air pump and circulating pumps and condensers. Inasmuch as the condensate from the paper machines could be pumped directly through an economizer to the boilers, we could get along without a feed heater. With little makeup water used, the condensate would be around 230–246 deg. F., before entering the economizer. The engine would cost not over 2½ times what the turbine cost, the electric-driven feed pump and electric drive for paper machines about 50 per cent more than steam-driven auxiliaries, and piping 50 per cent less. We can therefore safely assume that the new equipment can be installed for $88,000, or twice the former cost. This makes a total cost of $136,000 for the 600-lb. pressure plant.

Allowing 17 per cent for interest, taxes, insurance, maintenance and depreciation, the capital charge will be $23,120. The theoretical fuel cost will be 43.33 per cent of $79,275, or $34,352. But there are two things to consider in connection with generating steam at 600 lb. as compared with 190 lb. pressure. The temperature of saturated steam at 600 lb. being 487 deg. F., we will have a flue-gas temperature of approximately 590 deg. entering the economizer. If feed water entering the economizer at 240 deg. is heated to 340 deg., the final flue-gas temperature will be around 380 deg., which is 40 deg. lower than it is in the 190-lb. pressure plant and calls for about 3.5 per cent less fuel than if the same flue-gas temperature existed in the two plants. But to evaporate water from 240 deg. F. into steam at 600 lb. and 200 deg. superheat requires 1,337 — (240 — 32,) or 1,129 B.t.u., and in the first plant we had a feed temperature of 180 deg. and heat to produce a pound of steam at 190 lb. was 1,198.5 — (180 —32,) or 1,050.5 heat units. Thus $\frac{1,129 - 1,050.5}{1,050.5}$ or 7.47 per cent more heat is required per pound of steam for the high-pressure plant. Net increase 7.47 — 3.5, or 3.97 per cent, in addition to the 43.33 per cent. Allowing an extra fuel cost of 5 per cent above the first figure, $34,352, gives a total fuel cost of $36,069. Since we will burn only 48 per cent as much coal as formerly, the boiler-room labor can be reduced by one-half, making the cost $7,000 instead of $14,000, the present cost.

The total power cost with the proposed plant will be $23,120 + $36,069 + $7,000 = $66,189, or a difference of $107,995 — $66,189 = $41,806.

A saving of $41,806 per year could be made in a plant installed only two years ago, and yet that plant is considered modern and highly efficient! The main difficulty that an engineer would meet with in recommending that a 600-lb. pressure plant be designed and installed, when a new plant is being considered, is the total lack of precedent. The fact that a thing has never been done, seems to be the greatest obstacle to surmount. Some day we can expect to see such a power plant in operation, as has been described, but it will require a lot of development work. If power-equipment manufacturers were as alert and persistent as the central station advocates, industrial-plant managers would not so often fall for the apparent economy of central-station service. For all the advertising that superpower stations do about their record of "a kilowatt-hour per pound of coal," they have a condenser loss of over 50 per cent, and this loss must be passed along to the "power purchaser" instead of being eliminated, as it can be in a properly designed industrial plant of the class that is mentioned here.

Holyoke, Mass. ANDREW F. SHEEHAN.

[The relative advantages of high and low steam pressures is a live subject, but most of the discussion has centered about the application of high steam pressures in central stations. In the opinion of many engineers increased steam pressures will yield much larger returns in isolated plants, where the prime movers operate against the back pressure of a heating or process-steam system. Further discussion of this subject is invited. —EDITOR.]

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Finding Location for Spools of Duplex Pump

In putting new piston rods in a duplex pump, how is the location for the spools found? J. K.

After the pistons with the new rods have been replaced in the pump with the spools loose on the rods and with the cylinder heads in place, set each rod in the middle of its stroke. For this purpose move the rod as far as it will go when the steam piston is pressed hard against the cylinder head, first at one end and then the other end of the cylinder, and in each position make a mark on the rod at the end of a stuffing-box gland while the gland is held in the same position. Locate a point on the rod halfway between these marks and place the rod at tnat point of the stroke which will bring the middle mark even with the end of the gland from which the first marks were made.

Then with each rod standing at mid-stroke and with the rockers engaged with the spools loose on the piston rods, remove the steam-chest cover, place the steam valve of each side centrally over its ports and move the spools each to such a position on its rod that for each valve rod there will be the same amount of lost motion between the rod and its valve for movement of the valve rod in either direction. The spools then will be in the position where they should be secured to the piston rods and if the valve rods are of the proper lengths, the rocker arms will stand square with the piston rods. So that the pump may be started off by admission of steam to one end of one of the steam cylinders, before replacing the steam chest cover move one of the steam valves as far as it will go to one side of its central position.

Best Vacuum for Suction Lift

How does the suction lift of a pump affect the output in gallons? For example, having a 6-in. iron suction pipe 113 ft. long with one 90-deg. elbow with the pump 8 ft. 6 in. above the water in the well, and a vacuum gage connected to the suction chamber of the pump, how many inches of vacuum should it show for this lift? Also, what should be the reading of the vacuum gage if the pump was 17 ft. above the water in the well, and how many gallons will this suction pipe supply under these conditions? W. B.

The rate of flow through a water pipe increases when there is an increase of difference between the pressures at the inlet and outlet ends of the pipe. Consequently, when other conditions remain the same, lowering of the pressure—that is, increasing the number of inches of vacuum in the pump end of a suction pipe—must be accompanied by a higher rate of flow, and when the same vacuum is continued there must be a larger number of gallons of water handled per minute. However, the vacuum attainable is limited by liberation of air out of the water and vaporization of the water due to its temperature, and the speed of the pump must be regulated accordingly.

Under ordinary good working conditions, with the temperature of water 62 deg. F. and atmospheric pressure 14.7 lb. per sq.in., it is not found practicable to depend on obtaining a suction lift of over 22 ft. Since the pressure of the atmosphere would be the same as that exerted by a column of water 34 ft. high, it follows that when the water can be raised only 22 ft. the effective pressure of the atmosphere is reduced by 34 — 22 = 12 ft. head.

With a short suction pipe about 2 ft. water-column pressure is employed in overcoming friction at the entrance, in maintaining the velocity and in pipe friction, and the remainder, namely, 10 ft. of head, or 10 × 0.433 = 4.33 lb. per sq.in. absolute, is chargeable to vapor and air pressure, amounting to 30 — (4.33 ÷ 0.491) = 21.2 in. of vacuum. Higher vacuum in the pump suction chamber results in increase of head pressure, immediately followed by lowering of the vacuum.

Since there is 10 ft. of head destroyed by the imperfection of vacuum and about 2 ft. of head lost in pipe friction in the simplest form of suction pipe, it follows that when the suction lift is 8 ft. 6 in., the head available for overcoming the resistance to flow in a long horizontal suction pipe would be about 22 — 8.5 = 13.5 ft., which for flow in a 6-in. pipe 113 ft. long would be used up when the velocity of discharge is about 780 ft. per min. and with discharge of about 1,150 gal. of water per minute. However, the pump must handle the water without requiring a piston speed that would result in a poorer vacuum than 21.2 in. With ordinary pump proportions this vacuum would be attained when the suction-pipe velocity is only about 240 ft. per min., and to operate the pump at a speed that would obtain a higher vacuum would result in liberation of air and vapor, which would act as an obstruction to flow, and in this case higher vacuum would be quickly lost and result in less delivery.

If the lift is 17 ft., there would be 22 — 17 = 5 ft. head available for overcoming pipe resistances, and with that loss of head the velocity could be about 435 ft. per min. with delivery of about 640 gal. per min. But clearing this rate of delivery by the pump probably would result in excessive liberation of air and vapor, in average cases requiring, for a steady discharge, reduction of the speed to correspond with a suction-pipe velocity of 240 ft. per min., and the pump speed would have to be regulated to obtain no greater vacuum than 21 to 22 inches.

Required Size of Pop Safety Valve

How is the size of a pop safety valve, with 45-deg. bevel seat, determined for use on a power boiler? *J. B.*

According to the A.S.M.E. Boiler Code, the total relieving capacity of the safety valve or valves required should be determined on the basis of 6 lb. of steam per hour per sq.ft. of boiler heating surface for water-tube boilers; and for other types of power boilers, the minimum allowable relieving capacity should be determined on the basis of 5 lb. of steam per hour per sq.ft. of boiler heating surface, for boilers with maximum allowable working pressure above 100 lb. gage, and on the basis of 3 lb. of steam per hour per sq.ft. of boiler heating surface for boilers with maximum allowable working pressure at or below 100 lb. per sq.in.

The number of pounds of steam that a valve, with seat beveled 45 deg., will discharge per hour is given by the formula,

$$W = 110 \times P \times D \times L, \qquad (1)$$

where

W = Pounds of steam that the safety valve will handle per hour,

P = Absolute boiler pressure = gage pressure + 14.7 lb. per sq.in.

D = Inside diameter of valve seat, in.,

L = Vertical lift of valve disk, measured with 3 per cent excess pressure, in.

Knowing the total relieving capacity per hour required for the safety valve, according to the number of square feet of heating surface and type of boiler, and the working pressure, these quantities may be substituted in formula (1). For example, in a water-tube boiler having 2,200 square feet of heating surface, and which is to be operated at 185 lb. gage pressure, we would have

$W = 2{,}000 \times 6 = 12{,}000$ lb. of steam to be handled per hour,

$P = 185 + 14.7 = 199.7$ lb. per sq.in. absolute.

Substituting these values in (1) gives

$$12{,}000 = 110 \times 199.7 \times D \times L \qquad (2)$$

$$D \times L = \frac{12{,}000}{110 \times 199.7} = 0.54627 \qquad (3)$$

Allowing the lift L to be 0.13 in.,

$$D \times 0.13 = 0.54627 \text{ or } D = \frac{0.54627}{0.13} = 4.2 \text{ in. dia.}$$

The nearest commercial size of valve suitable would be 4½ in. diameter.

Efficiency of Butt and Double-Strap Joint Double-Riveted

What would be the efficiency of a butt and double-strap joint, double-riveted, where the thickness of the main plate is ⅜ in., thickness of butt straps ⅜ in., tensile strength of plates 55,000 lb. per sq.in. cross-section, pitch of rivets 4¾ in., diameter of rivet holes ⅞ in.? *R. M.*

The efficiency would be the least strength for a unit of pitch P as shown in the figure, divided by the strength of an equal length of solid plate. Having a tensile strength of plate equal to 55,000 lb. per sq.in. of cross-section, and thickness of plate ⅜ in. = 0.375,

(1) The strength of solid plate per unit of pitch P = 4¾ in., would be

$$4.75 \times 0.375 \times 55{,}000 = 97{,}969 \text{ lb.}$$

Having thickness of butt straps ⅜ = 0.3125 in., diameter of rivets after driving, ⅞ = 0.8125 in., giving cross-sectional area of rivets = $0.8125 \times 0.8125 \times 0.7854 = 0.5185$ sq.in., and allowing strength of rivets in single shear = 44,000 lb. per sq.in. of cross-section and for double shear = 88,000 lb. per sq.in. of cross-section; also allowing the crushing strength of material to be 95,000 lb. per sq.in., the strength of joint for a unit of pitch P would depend on the least value obtained from the following considerations:

(2) Strength of plate between rivet holes in the outer row,

$$(4.75 - 0.8125)\ 0.375 \times 55{,}000 = 81{,}211 \text{ lb.}$$

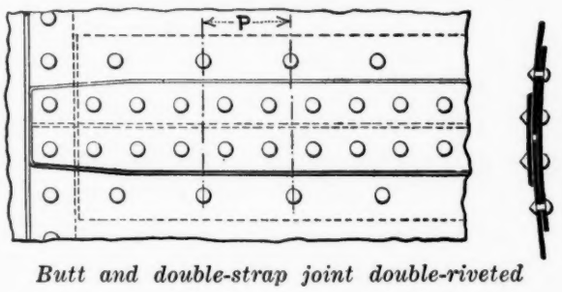
(3) Shearing strength of two rivets in double shear, plus the shearing strength of one rivet in single shear,

$$2 \times 88{,}000 \times 0.5185 + 1 \times 44{,}000 \times 0.5185 = 114{,}070 \text{ lb.}$$

(4) Strength of plate between rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row,

$$(4.75 - 2 \times 0.8125) \times 55{,}000 + 1 \times 44{,}000 \times 0.5185 = 194{,}689 \text{ lb.}$$

(5) Strength of plate between rivet holes in the second



Butt and double-strap joint double-riveted

row, plus the crushing strength of butt strap in front of one rivet in the outer row,

$$(4.75 - 2 \times 0.8125)\ 0.375 \times 55{,}000 + 0.8125 \times 0.3125 \times 95{,}000 = 88{,}574 \text{ lb.}$$

(6) Crushing strength of plate in front of two rivets, plus the crushing strength of butt strap in front of one rivet,

$$2 \times 0.8125 \times 0.375 \times 95{,}000 + 1 \times 0.8125 \times 0.3125 \times 95{,}000 = 82{,}012$$

(7) Crushing strength of plate in front of two rivets, plus the shearing strength of one rivet in single shear,

$$2 \times 0.8125 \times 0.375 \times 95{,}000 + 1 \times 44{,}000 \times 0.5185 = 80{,}704 \text{ lb.}$$

Failure of butt straps between rivet holes in the inner row is not possible on account of the relative thicknesses of main plates and butt straps.

The least strength would be obtained from consideration (7), namely 80,704 lb. per unit of pitch, and the efficiency of the joint would be 80,704 ÷ 97,969 = 0.823, or practically 82 per cent.

Relief Pressure of Superheater Safety Valve

Why is it necessary for a superheater safety valve to be set to blow at a lower pressure than the main safety valve of the boiler? *R. L. D.*

This is essential to provide a flow of steam through the superheater, and thus prevent burning out, in case the load should suddenly be thrown off the boiler or plant.

[Correspondents sending us inquiries, should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications, and for the inquiries to receive attention.—Editor.]

Here and There in the Power Plant

Sidelights on things generally used but less generally understood

Area, Volume and Flow

There are two simple formulas that have a very wide application in power-plant engineering. These are:

(1) Cross-sectional area × length = volume.

(2) Volume × density = weight.

The first formula applies to any solid body whose cross-section is constant. It also applies to liquids, gases and vapors, as will be shown presently.

Taking up first the case of solids, the simplest to consider is a square rod. If the rod is one inch square and seven inches long, the volume is evidently 7 cu.in. If the rod is two inches square and seven long the cross-sectional area is 4 sq.in. Each inch of length is equivalent to four one-inch cubes, the total number for the entire solid being $4 \times 7 = 28$. Hence in this case also the volume equals the product of the length and the cross-sectional area.

The formula for a cylinder is:

$$Volume = 0.7854 \times (diam.)^2 \times length.$$

Here it is seen that the first two quantities on the right side of the equation simply give the area of the circle that forms the cross-section of the cylinder, so the rule given at the start evidently fits this case also.

To understand the more general application, it will help to picture the action of the extrusion machine, which is widely used for manufacturing rods, turbine blading and other sections in such alloys as brass and bronze.

The Extrusion Machine

A heavy die is made, the hole in which has the shape desired in the finished product (allowing a little for the finishing draw). This die is clamped on the end of a cylinder made by boring a large hole (6 to 8 in. in diameter) through a heavy steel forging. In this cylinder is placed the billet of brass or other alloy after it has been heated to a plastic condition. When the billet is compressed by a hydraulically operated plunger, the metal is forced out through the die in the form of a rod of the desired section, much as tooth paste is forced from a tube. If the extruded shape is a square or round rod, it is evident from what has been said that its volume is the product of the length by the cross-sectional area. But suppose the shape is more complex—that of turbine blading for example—does the rule still hold? Without going into the theory, it may be definitely stated that, for any shape that could conceivably be extruded, the volume will be equal to the cross-sectional area times the length. Expressed in another way the volume is equal to the product of the cross-sectional area by the length for any straight solid with a uniform cross-section of any shape whatever.

Practical Applications

Suppose it is desired to find the volume of metal in a bushing of 2-in. inside diameter, 2.5-in. outside diameter, and 5-in. length, the bushing being a straight shell without a flange. The metal surface of the cross-section forms a ring whose area is the difference between the area of a 2.5-in. circle and that of a 2-in. circle. This difference is 1.7671 sq.-in. Multiplying this by the length, 5 in., gives a volume of 8.835 cu.in. If the material used is bronze weighing 0.32 lb. per cubic inch, the weight of the bushing will be $8.835 \times 0.32 = 2.83$ lb. In this last step we have really applied the second formula.

Care must be taken to use corresponding units throughout. If the area is measured in square inches and the length in inches, the volume will be in cubic inches. If, further, the density is given in pounds per cubic inch, the weight found will be in pounds. Correspondingly, the use of square feet, feet and pounds per cubic foot will give the volume in cubic feet and the weight in pounds.

Flow of Water in Pipe

The simplest case of a fluid to consider is that of the flow of water through a pipe.

Problem: With a velocity of 10 ft. per second how many gallons of water per hour will be delivered by a 2-in. standard-welded pipe?

The internal diameter is 2.067 in. and the cross-sectional area of the opening 3.356 sq.in. Water passes any given point of the pipe exactly as the metal flows from the die of an extrusion machine. In one second a cylinder of water 10 ft. (120 in.) long passes a given point. It is helpful to picture this as an actual cylinder. The volume of this cylinder will be $120 \times 3.356 = 402.72$ cu.in. Since there are 231 cu.in. in one gallon, the cylinder will contain $402.72 \div 231 = 1.743$ gallons. Since this is the amount of water that passes a given point each second, the flow per hour will be $1.743 \times 3{,}600 = 6{,}275$ gallons.

To reverse this type of problem, suppose it is desired to find the size of pipe that will deliver 500 lb. of water per minute with a velocity of flow of 15 ft. per second. The water passing a given point in one minute can be represented by a cylinder $15 \times 60 = 900$ ft. long and weighing 500 lb. The first step is to compute the volume of this cylinder, which will be $500 \div 62.5 = 8.00$ cu.ft., 62.5 lb. being the approximate weight of one cubic foot of water at room temperature.

Since volume equals cross-sectional area times length, the cross-sectional area must be equal to the volume divided by the length. In this case it will be $8 \div 900 = 0.00889$ sq.ft. or $0.00889 \times 144 = 1.28$ sq.in. A one-inch pipe has an internal cross-sectional area of 0.864 sq.in. and an 1¼-in. pipe, an area of 1.496 sq.in., so the latter size would meet the requirements.

Flow of Steam

Problems involving steam flow in pipes can be solved in a similar manner, the weight per cubic foot for the given pressure and superheat being taken from the steam tables.

Suppose a 200-kw. condensing turbine consumes at full load 19 lb. of steam per kilowatt-hour, steam being supplied at 150 lb. gage pressure and 100 deg. superheat. It is desired to pick a suitable size for the steam line, assuming a permissible steam velocity of 3,000 ft. per min. The volume of one pound of steam under the given conditions is 3.21 cu.ft. Therefore the total hourly supply of steam to the turbine at full load will be $200 \times 19 \times 3.21 = 12{,}198$ cu.ft., or $\div 60 = 203$ cu.ft. per minute. The problem then is to find the cross-sectional area of a cylinder 3,000 ft. long and containing 203 cu.ft. This will be $203 \div 3{,}000 = 0.0677$ sq.ft. or $0.0677 \times 144 = 9.75$ sq.in. A 4-in extra-heavy pipe, with a cross-sectional area of 11.5 sq.in. would meet the requirements.

Discharge of Fan

As an another example consider the case of a ventilating fan with a discharge area of 3 sq.ft. Measurements by pitot tube or anemometer show the average air velocity over the entire area of the discharge to be 4,000 ft. per minute. The volume of air delivered will then be $4{,}000 \times 3 = 12{,}000$ cu.ft. per minute.

In dealing with the flow of fluids through openings, always use the average velocity. The actual velocity varies, being greatest at the center and least at the outside.

Other examples of a similar nature could be given, but there is space only to call attention to the fact that the plunger of a pump (neglecting slip) delivers at each stroke a cylinder of water whose length is the length of the stroke and whose diameter is the bore of the pump. Dividing this volume in cubic inches by 231 (the number of cubic inches in a gallon), and multiplying by the number of working strokes per minute, gives the pump delivery in gallons per minute.

Lap-Seam Boiler Explosion Kills Three

Examination Reveals Crack Extending from Head to Head

A BOILER in Watt's Cheese Factory at Bishop Mills, Ontario, about 47 miles southwest of Ottawa, exploded on the morning of Aug. 12, killing the owner and two others and injuring three.

The boiler was a 30-in. diameter, 7-ft. return-tubular type about eight years old. It was made of ¼-in. plate of one course but in two sheets, with a single-riveted lap seam on each side about the center line. The rivet holes were $\frac{13}{16}$ in. in diameter and 2-in. pitch, having been punched full size. The heads were ½ in. thick and a 20-in. diameter drum, having a ¼-in. flat and unstayed head, remained intact although only single-riveted to the shell.

The explosion almost completely demolished the factory, and the boiler landed in a field about 80 yards distant, the tubes being scattered over a considerable area. Up to this writing the steam gage and safety valve had not been found. The safety valve was said to have been set to blow at 90 lb., but this could not be confirmed.

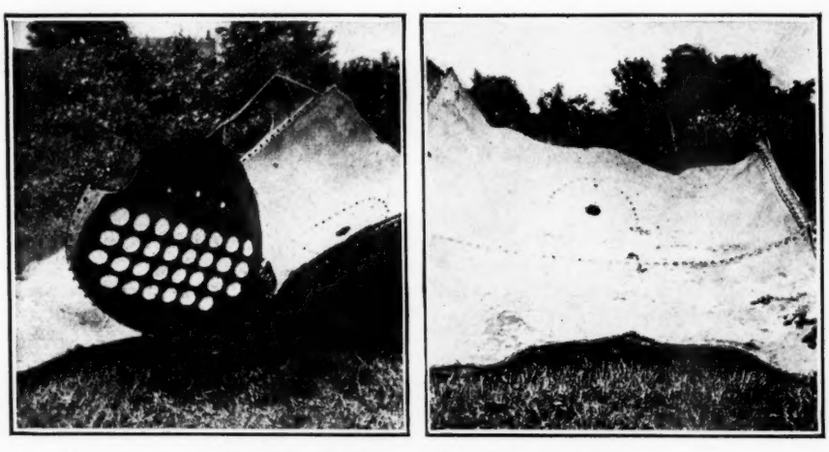
The shell ruptured about 1½ in. from the longitudinal seam rivets, tearing from head to head. At the rupture a crack was plainly visible completely through the full thickness of the plate for a distance of about 4 in. near the center, tapering off to sound metal some 6 in. from each head. Excepting these sound portions, the average thickness of good metal in the length of the seam was about $\frac{1}{16}$ in. For about 21 in. the crack extended through within $\frac{1}{32}$ in. of the outer edge, constituting a weakness quite sufficient to cause failure at normal pressure.

On the other side of the rivets heavy grooving had occurred to a depth of about ⅛ in. With the exception of this grooving and the lap-seam crack the plates were in good condition and not affected by corrosion to any appreciable extent.

The boiler was last inspected four years ago, at which time it was pronounced in good condition by a local engineer. This same engineer, testifying at the inquest, gave as his opinion that the accident had been caused by low water followed by the injection of cold feed, and the verdict of the jury was rendered accordingly. However, an examination of the plates failed to show any signs of overheating, and it is quite evident that this accident should be recorded as another lap seam failure.

Estimate of Tokyo's Light Plant Loss Given

The *Times* (London) recently stated that the cost of the damage caused to the plant of the Tokyo Electric Light Co. by the recent serious earthquake is estimated at $9,021,250. In addition to repairing this damage the company is undertaking improvements that will cost a further $3,500,000. The necessary funds will be obtained partly from reserves and partly from the reduction of the ordinary dividend for a few years. Seven of the company's 30 hydro-electric power stations were damaged, but in all cases the generating machinery escaped injury. The four steam stations, representing 10 per cent of the generating capacity, were damaged. Thirty-seven of the 111 substations were damaged, but only 13 were totally destroyed. With the exception of about a quarter of the overhead equipment the transmission system, it is reported, virtually escaped damage.



Views of shell showing line of failure

Beaver Station Power Plant Plans Being Drawn

The Duquesne Light Co., of Pittsburgh, Pa., is having plans drawn for a new power plant to be known as the Beaver Station at Shippingsport on the Ohio River. It is understood that the initial installation will contain two 35,000-kw. units.

Success Assured for Power Show

The Third National Exposition of Power and Mechanical Engineering, which is to be held at the Grand Central Palace, New York City, Dec. 1-6, has had twice the requests for space that the previous exposition had at a corresponding time last year. The attendance last year increased 30 per cent over that of the year before, and engineers and manufacturers all over the country are planning to be present at the coming event. This is definite proof that the show will be a success.

Plans for Susquehanna River Project Progressing

The Susquehanna Power Co. will present an application for license to the Federal Power Commission, in the near future, covering its 360,000-hp. development on the Susquehanna River at Conawingo, Pa. The preparation of the plans for the license has been entrusted to Jackson & Moreland, of Boston.

The power is to be supplied to the Philadelphia Electric Co. Since this project will develop at least 80 per cent as much power as has been harnessed on the American side of Niagara Falls, its progress is being followed with wide interest.

Large Steam Plant Planned for Broad River

An option has been taken on the Public Utility power development in Columbia, S. C., and on the Parr Shoals project on Broad River by the General Gas & Electric Corp. The company now is studying the feasibility of installing a large steam plant on Broad River to operate in connection with the steam plant in Columbia and the hydro-electric development.

The plan also includes, it is understood, the consolidation of various isolated plants and the building up of a strong system. The territory tributary to Columbia is recognized as one having material potentialities.

Highest-Head Reaction Turbine Placed in Operation

The highest-head reaction turbine yet constructed went into operation, when on Aug. 4, 1924, the first unit of the Oak Grove hydro-electric development of the Portland Electric Power Co. was placed in operation. The unit is a Pelton, 35,000-hp. 514-r.p.m. vertical turbine operating under an effective head of 860 ft., and a 30,000-kva. 11,000-volt 60-cycle G. E. generator, with direct-connected exciter.

Water for the plant is diverted from the Oak Grove Fork of the Clackamas River and is carried a distance of 6.6 miles through a 9-ft. steel-pipe flow line to a surge chamber above the power house. The penstock is 1,360 ft. long, 8 ft. in diameter at the top and 6 ft. at the bottom. At the top of the penstock is a 66-in. Johnson valve which can be operated from the power house and which will close automatically in 30 seconds in case of a break in the penstock. At the bottom is a 72-in. butterfly valve which is a part of the waterwheel casing.

The first unit of the Oak Grove development which is about 50 miles southeast of the City of Portland, Ore., was completed after three years of construction, at a cost of $8,000,000. The ultimate capacity of the development will be 75,000 kw., and more units will be added as demands for power require. A detailed description of the first unit with some of the interesting features of the prime mover and the development will be published in an early issue of *Power*.

Oak Grove Power House

California Pipe Line Buys New Pumping Units

A large order was recently placed by the Pan-American Petroleum Co., of Los Angeles, with the Wilson-Snyder Manufacturing Co., of Pittsburgh, Pa., for ten pipe-line pumps for its California field. Each pump is to have a capacity of 15,000 bbl. per day against 750 lb. per sq.in. line pressure and is to be driven by a 300-hp. Worthington Diesel engine.

French Conference Will Be Held in June, 1925

The Conference Internationale des Grands Réseaux Electriques à Haute Tension will hold its third session toward the end of June, 1925, and will extend its sessions over a week. The conference has for its objects the study of all problems relating to the production of electrical energy in large steam or hydraulic stations, the interconnection of large power stations, the construction of high-tension networks on a large scale, and the operation of such networks. Visits to works and pleasure trips will be arranged in connection with the conference. M. J. Tribot Laspiere, Boulevard Malesherbes 25, Paris, can furnish any other information that is desired.

Giant Power Advisory Board Membership

The persons invited by Governor Pinchot for membership on the Giant Power Advisory Board of which Maj. Gen. William Crozier, former chief of the Ordnance Department of the Army, is chairman, who have accepted service, are:

Maj. Gen. William Crozier, U.S.A., Riggs Nat'l Bank, Washington.

Arthur E. Morgan, Antioch College, Yellow Springs, Ohio.

Francis Lee Stuart, 949 Broadway, New York City.

William S. Twining, 119 Harvey St., Germantown, Pa.

Fred R. Low, Editor of *Power*, 10th Ave. at 36th St. New York City.

Joseph Hyde Pratt, 6 Government St., Asheville, N. C.

Irving Fisher, Yale University, New Haven, Conn.

Charles E. Merriam, University of Chicago, Chicago, Ill.

George Foster Peabody, Yaddo, Saratoga Springs, N. Y.

George W. Norris, Federal Reserve Bank, Philadelphia, Pa.

Herbert Quick, Berkeley Springs, W. Va.

Leo S. Rowe, Pan American Union, Washington, D. C.

Admiral Casper F. Goodrich, Princeton, N. J.

Samuel Gompers, A. F. of L., 9th and Mass. Ave., Washington, D. C.

Herbert Knox Smith, Hartford Empire Co., Hartford, Conn.

Arthur F. Davis, East Bay Municipal Utilities Dist., Oakland, Calif.

Charles W. Eliot, Pres. Emeritus of Harvard, Cambridge, Mass.

Joseph N. Teal, Spaulding Bldg., Portland, Ore.

William Allen White, Emporia, Kan.

Fred A. Gaby, Hydro Power Com. of Ontario, 190 University Ave., Toronto.

John R. Freeman, 815 Grosvenor Bldg., Providence, R. I.

Henry L. Stimson, 32 Liberty St., New York City.

Arthur J. Mason, 5715 Woodlawn Ave., Chicago, Ill.

Raymond B. Stevens, Landaff, N. H.

William Kent, Kentfield, Calif.

Henry S. Graves, Yale University, New Haven, Conn.

Mrs. Martha Brisley Bruere, *Survey*, 112 E. 19th St., New York City.

Henry A. Garfield, Williams College, Williamstown, Mass.

William B. Mayo, Care of Henry Ford, Dearborn, Mich.

James R. Garfield, Nat'l City Bank Bldg., Cleveland, Ohio.

William Mulholland, Los Angeles, Calif.

Charles H. Thompson, International Gypsum Corp., Engineers' Club, Boston, Mass.

Mrs. Maud Wood Park, 532 17th St., Washington, N. W., D. C.

The Board will be asked to pass on the final report of the survey before it is submitted to the Pennsylvania Legislature. The report will not be ready, it is reported, for a year or two.

The Dominion Government has voted $15,000 toward the expenses of the World Power Conference.

Canada Leads in per Capita Use of Electricity

Statistics just compiled by the Dominion of Canada's Bureau of Statistics show that Canada leads the world in the per capita consumption of electric energy. Each inhabitant consumes 820 kw.-hr. of electrical energy in a year. There is an electric meter installed in Canada for every five inhabitants and 454 hp. is installed for every 1,000 population. The present recorded waterpower resources of Canada will permit of a turbine installation of 41,700,000 hp., of which today only 3,250,000 is installed.

Columbia University Modifies Engineering Courses

According to recent announcement by Dean George B. Pegram, there have been some modifications adopted by the School of Mines, Engineering and Chemistry, of Columbia University which make it possible for the student of these courses to obtain a degree in five instead of six years. Pre-engineering training may be completed in two years instead of three.

A student may now enter Columbia College with credits in elementary physics and chemistry and advanced mathematics and on completion of a prescribed course, covering only two years and a summer session, be admitted to the School of Mines, Engineering and Chemistry.

Hitherto the shortest possible time for completing the pre-engineering preparation has been three years, hence the new arrangement will make it possible for a student to be graduated with an engineering degree in five instead of six years. As compared with the regular pre-engineering course, this shortened course is somewhat heavier in hours per week and contains exactly the same scientific studies and approximately one-half year's less work in modern language and general elective subjects. The regular three-year pre-engineering and three year engineering courses are retained, however.

Students taking the regular three-year pre-engineering course exercise the professional option and receive the A.B. degree on the completion of the first year in the Engineering school. Students taking the shortened pre-engineering course will not be eligible for the A.B. degree under the professional option, and the faculty has therefore voted to award the B.S. degree to these students at the end of their fourth year, that is, at the end of the second year of the regular three-year professional course in engineering. Thus a student taking this course will not be penalized by having to wait five years before receiving his first degree.

The Rochester Gas and Electric Co., Rochester, N. Y., is remodeling and extending the capacity of its power plant by the installation of three boilers with 8,750 sq.ft. of heating surface, fired by Taylor stokers with steam and ash dump.

Equipment for Steam Station Rushed to Relieve California Power Shortage

A SPECIAL train of 22 carloads of steam and electrical equipment, valued at approximately a million dollars, left the Schenectady Works of the General Electric Co. Saturday night, August 3, nine weeks ahead of schedule as a contribution to combating the water power shortage in California.

This trainload of equipment includes a complete turbine-generator unit, together with many items of accessories, for the 100,000-kw. steam power station that is being rushed to completion for the Southern California Edison Co.

The orders for the electrical equipment were placed in January. The trainload that left Saturday included: One 35,000-kw. turbine-generator; one 2,800-kw. house generator; one 45-kw. direct connected exciter; two motor-operated field rheostats for main and auxiliary generators; five 100-hp. motors; one solenoid-operated field rheostat for exciter; five 150-hp. motors; three 73,000-volt 400-amp., oil circuit breakers; six 13,000 kva., 72,000/4,100/600/11,000-volt oil cooled transformers; one 350-kw. motor-generator set and other auxiliary equipment.

Under ordinary circumstances this shipment would be of interest chiefly because it represents the largest single generating unit shipped west of the Rocky Mountains. This fact, however, is eclipsed by the way in which the order was rushed from foundry to final tests.

The acute power situation which made this shipment so urgent is caused by one of the greatest water shortages in the history of California. The unprecedented drought has resulted not only in a shortage of power, but also in an increased power demand by farmers for irrigation pumping.

The Southern California Edison Co. has already placed in service a total of 26,620 hp. steam equipment. In addition, a record installation was made of a 13,000 hp. turbine and also of two 6,500-hp. machines.

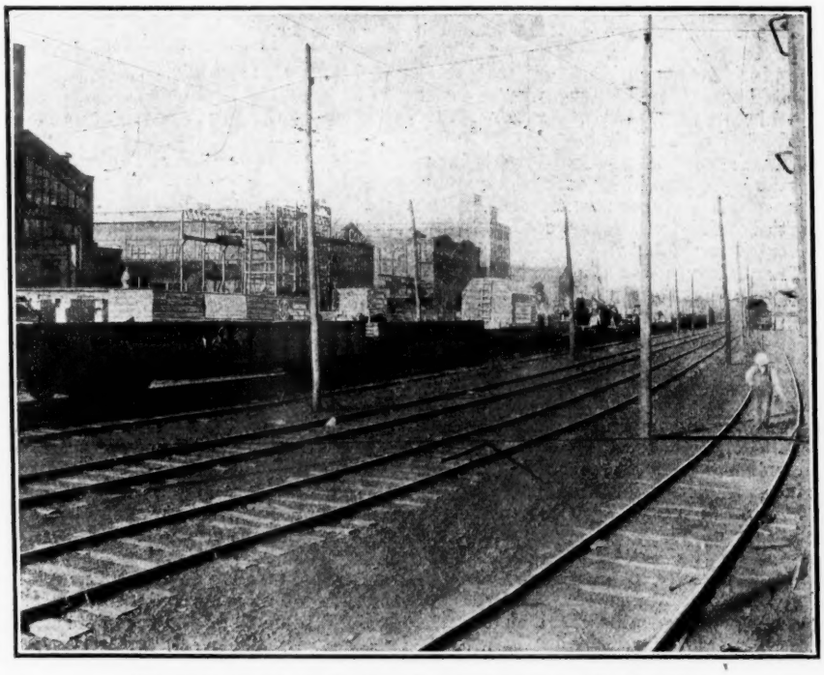
The new steam plant at Power, Calif., which will be equipped with the apparatus just shipped, is expected to do much to relieve the situation, when the apparatus is assembled, tested and ready for operation.

Peru Good Market for Power Machinery

The growing importance of Peru as a market for American machinery is shown by the consistent gain in position among machinery markets of the world from twenty-first in 1910 to fourteenth place in 1923. Peru now outranks British South Africa and Spain and is next to the Philippines in importance, according to figures published by the Department of Foreign and Domestic Commerce. The following figures from United States custom returns give exports of industrial machinery from the United States to Peru in 1922 and 1923:

Class	1922	1923
Power generating machinery, except electric	$205,468	$257,007
Construction and conveying machinery	58,659	120,710
Mining, oil-well, and pumping machinery	468,460	912,286
Miscellaneous other machinery except agricultural	354,367	575,928

American goods lead in the Peruvian market. Conditions are stable, and there is reason to suppose that machinery sales will continue to show gradual gains.



View of special train carrying equipment to California to relieve power shortage

New Hazard in Powdered Coal Installations

A new hazard in the use of powered coal (carbon-monoxide poisoning) has been brought to the attention of engineers and plant executives through an accident in a cement plant. The accident, which is related in *The Travelers Standard*, was:

"The accident to which we refer occurred a few months ago in a cement plant in Alabama. At this plant there are three bins for storing powdered coal. Each bin is 14 ft. square at the top and the bottom is hopper-shaped, so that the coal will flow down by gravity into a screw conveyor, which feeds it to the blowers of the cement kilns. In one corner of the cover of each bin there is an opening 30 in. square, which is provided with a door, and at the opposite corner of each bin there is a vent opening 10 in. in diameter. The coal is fed to the bins after being thoroughly dried, and it is sometimes quite hot when so deliverd.

"At the time of the accident two men had been detailed to clean off some powdered coal that had stuck to the sides of the bins. One of these men went down into a bin, and shortly afterward his fellow employee saw him fall over apparently unconscious. The second man called for assistance and immediately went down into the bin to help his companion, but was also rendered unconscious. Neither of these men had ropes secured to them when they entered the bin. A third man was provided with a gas-mask of some kind, a rope attached to his body, and he was lowered into the bin to attempt a rescue. He succeeded in making a rope fast to the legs of one of the unconscious men, but he was also overcome. Still another volunteer tied a wet handkerchief tightly over his nose and mouth, took a long breath, and was lowered into the bin by means of a rope. He also lost consciousness, but before doing so was able to attach a rope to the first man who went into the bin. Eventually, all four men were hoisted out. The first two men to enter the bin died, but the other two recovered after a short time.

"There can be no doubt that these deaths were caused by carbon-monoxide gas, which had been generated by the partial oxidation of the coal and had collected in the bin."

The article in *The Travelers Standard* suggests that to prevent such accidents from carbon-monoxide poisoning, the bins or other inclosures should be cleaned by means of special tools, but if absolutely necessary that they be cleaned by hand, the inclosure should be thoroughly ventilated before a man enters it; that any man entering such an inclosure should wear a self-contained oxygen-breathing apparatus or a hose-mask supplied with an abundant quantity of fresh air by means of a pump, or its equivalent or a gas-mask especially designed for protection against carbon monoxide. Ordinary gas-masks, respirators and cloths tied over the face afford absolutely no protection against carbon monoxide, which is both colorless and odorless and gives no warning of its presence.

Mining Industry Exposition in Peru in December

American manufacturers of mining machinery, pumps, etc., have been invited to exhibit their products at the Peruvian Mining Industry Exposition to be held at Lima during the month of December of this year, according to *Commerce Reports*. A special section of the exposition will be devoted to mining and metallurgical machinery, all space being obtainable gratis. Correspondence should be addressed to the Direccion de Minas, Ministerio de Fomento, Lima, Peru.

Winnipeg Plans Hydro Extension

According to recent press reports, extension of hydro-electric service to all towns in Manitoba within serving distance of the power lines and the formation of a hydro league along the lines of the Ontario Power Commission is the aim of J. G. Glassco, manager of the Winnipeg Hydro System. This system is owned by the City of Winnipeg, and at the present time supplies power for industrial and domestic purposes only to that city at rates that are the lowest on this continent.

At a meeting of the utilities committee of the city council, Mr. Glassco sought permission to hold a special excursion to Point du Bois, where the hydro system's plant is located, in order to start a movement for the formation of a hydro league. He stated that a demand was growing up in the rural centers for hydro service, and in his opinion the organization of a league something like that of the Ontario Power Commission should be the first move toward extending the work of the hydro system.

At the present time, negotiations are in progress for the extension of the hydro's service to the City of Brandon which is the second largest in the province.

Austria Developing Her Water Power

Austria lacks today food and coal; her own land can feed her for only three months of the year, and she produces only 30 per cent of the coal she uses. She is, therefore, forced to import both food and coal in enormous quantities, and she will always be dependent on foreign countries for her food supplies, but she is trying to free herself as rapidly as possible in respect to her fuel. And the solution of this second problem lies in her water power.

She is already putting into use 13 per cent of her total resources, with an additional 4 per cent under development. When the whole of the water power is available, it will reduce her coal requirements to one-half and her imports of foreign coal to one-third. In 1922 some 6,000,000 tons of coal and coke were imported at an approximate cost of $57,000,000.

In 1923 there were about 1,000 water-power plants still under construction, with an expected aggregate maximum output of some 265,000 hp., and this is in itself almost a half of the whole of the previous output, which was 582,000 hp. It is estimated that the plants now being constructed will alone render possible the diminution of coal consumption by about 1,000,000 tons yearly, thereby reducing Austria's future liabilities to foreign countries by approximately $10,000,000, according to press reports.

There were sold, according to figures published by the Department of Commerce, during July, 115 stokers with aggregate capacity of 37,759 hp., which were installed under 14 fire-tube boilers having 16,600 sq.ft. of heating surface and 101 water-tube boilers having 360,990 sq.ft. of heating surface.

Water-Power Projects

Connecticut River Project Abandoned.—The Connecticut River Power Co., which has held a preliminary permit for nearly three years, for a project Windsor Locks, Conn., where 50,000 hp. is available, has notified the Federal Power Commission that it will not be able to apply for a license. It was found that the old vested rights along the river, together with the cost involved in providing for navigation, constituted obstacles that could not be overcome at this time.

Federal Power Commission To Make Important Decision.—Because of a question raised in connection with the Electra plant of the Pacific Gas & Electric Co. on the Mokelumne River in California, the Federal Power Commission must render a decision as to the exact percentage of public lands in the project area which is to determine whether or not a major license must be issued.

The Electra plant has been in operation 14 years. The company contends that the public lands involved were put to this use before legislation was passed requiring Federal rights. It is willing, however, to be put under that type of license which covers a minor part of a project which does not require special accounting to the Federal government. The company is of the opinion that were the case taken to court a decision would be forthcoming which would free it from taking out any license, but to avoid litigation it is willing to operate under a minor license.

In cases previously passed upon the Commission has made no effort to determine the exact line of demarcation between major and minor licenses, but the inclination apparently has been to regard a major license necessary when public lands exceed 15 per cent of the project area.

occupy approximately equal portions

The structures of the Electra project of public and private lands, but if the reservoirs are considered the proportion of public lands is reduced to 10 per cent. In this particular instance the public interest is not affected, since the plant for many years has been under the jurisdiction of the state, but the

Commission's decision in this case will be of significance because of its bearing on future applications.

Society Affairs

The National Safety Congress, which is to be held at Louisville, Ky., Sept. 29-Oct. 3 will have several subjects of interest to power plant men. On the Ice and Refrigeration Section program there will be a discussion of "Plant Hazards"; The Public Utilities Section will have a paper by Homer Niesz on "Instruction and Observance of Safety Rules in Station and Sub-Station Operation."

Personal Mention

Dr. Johann Stumpf, Professor der Technischen Hochshule at Charlottenburg, Germany, and inventor of the unaflow engine, is now visiting in the United States.

Robert Thurston Kent, a well-known mechanical engineer, has been appointed to the newly created position of Superintendent of Prison Industries of New York State, at a salary of $7,500 per year. The appointment became effective August 20, when Mr. Kent returned from the Prague International Management Congress, held under the auspices of the Czecheslovakian government, where he represented the American Engineering Council.

Business Notes

The Universal Coal Spreader Co., 53 West Jackson Blvd., Chicago, announces the appointment of Memmer & Kimball, Syndicate Trust Bldg., St. Louis, as sales agents for southern Illinois and eastern Missouri.

The McClave-Brooks Co., Scranton, Pa., announces that on Sept. 1 it will discontinue its Boston office and that all of the New England territory will come under the supervision directly of the New York office.

The William Cramp & Sons Ship & Engine Building Co., Philadelphia, Pa. and **The Pelton Water Wheel Co.,** San Francisco and New York, announce the opening of a Southern office in the American Trust Bldg., Birmingham, Ala. Quin W. Stuart will be the district representative.

Trade Catalogs

Heaters, Buffalo Forge Co. Buffalo, N. Y. Leaflet describing "Breezo-Fin" heaters and thermostat for automatic room temperatures control.

Drills—Cleveland Twist Drill Co., Cleveland, Ohio. Booklet containing speed and feed tests made at the American Railway Association convention at Atlantic City June 11-18.

Coming Conventions

American Ceramic Society. Ross C. Purdy, Lord Hall, Ohio State University, Columbus Ohio. Convention at Los Angeles, Calif., Oct. 6-7.

American Chemical Society. Dr. Charles L. Parsons, 1709 G St., N. W., Washington, D. C. Sixty eighth meeting at Cornell University, Ithaca, N. Y. Sept. 8-13.

American Electric Railway Association. James W. Welsh, 8 West 40th St., New York City. Meeting at Atlantic City, Oct. 6-10.

American Electrochemical Society. Dr. Colvin G. Fink, Columbia University, New York City. General meeting at Detroit, Mich., Oct. 2-4.

American Institute of Electrical Engineers. F. L. Hutchinson, 29 West 39th St., New York City. Fall convention at Pasadena, Calif., Oct. 13-17.

American Institute of Mining and Metallurgical Engineers. F. F. Sharpless, 29 West 39th St., New York City. Autumn meeting at Birmingham, Ala., Oct. 13-15.

American Society of Civil Engineers. 29 West 39th St., New York City. Fall meeting at Detroit, Oct. 23-25.

American Society of Mechanical Engineers. Calvin W. Rice, 29 West 39th St., New York City. Annual meeting at New York City, Dec. 1-4.

American Society of Refrigerating Engineers. W. H. Ross, 35 Warren St., New York City. Annual meeting at New York City, Dec. 1-3.

Association of Edison Illuminating Companies. P. S. Millar, 80th St. and East End Ave., New York City. Meeting at Griswold Hotel, New London, Conn., Sept. 8-12.

Association of Iron & Steel Electrical Engineers, John F. Kelly, 1007 Empire Bldg., Pittsburgh, Pa. Annual meeting and exposition at Duquesne Garden, Pittsburgh, Pa., Sept. 15-20.

Eastern Ice Manufacturers Association. W. H. Ross, 35 Warren St., New York City. Eighteenth annual meeting at Hotel Chelsea, Atlantic City, N. J., Nov. 12-14.

Franklin Institute of Pennsylvania, Centenary Celebration at Philadelphia, Sept. 17-19.

Illuminating Engineering Society. Norman D. MacDonald, 29 West 39th St., New York City. Convention at Briarcliff Lodge, Briarcliff Manor, N. Y., Oct. 27-31.

International Union of Steam and Operating Engineers. Dave Evans, 6334 Yale Ave., Chicago, Ill. Biennial convention at Detroit, Mich., Sept. 8-13.

National Association of Practical Refrigerating Engineers. E. H. Fox, 5707 West Lake St., Chicago, Ill. Convention at New Orleans, La., Nov. 11-14. Exhibits at the Isaac Delgado High School.

National Association of Stationary Engineers. Fred W. Raven, 417 South Dearborn St., Chicago, Ill. Annual convention and exhibition at Hotel Pantlind, Grand Rapids, Mich., Sept. 8-13. Annual conventions and exhibitions of the state associations are scheduled as follows: **Michigan Association** at Grand Rapids, Sept. 8. Wm. H. Yeomans, 209 Plainfield Ave., Grand Rapids. **Pennsylvania State Association** at Grand Rapids, Mich., Sept. 8. J. N. Calvert, Union & Ridenour Ave., Crafton, Pittsburgh, Pa.

National Exposition of Power and Mechanical Engineering. C. F. Roth, Grand Central Palace, New York City. Annual exposition at New York City, Grand Central Palace, Dec. 1-6.

National Safety Council, W. H. Cameron, 168 N. Michigan Ave., Chicago, Ill. Thirteenth Annual Safety Congress at Louisville, Ky., Sept. 29 to Oct. 3.

New England Water-Works Association, Frank T. Gifford, 715 Tremont Temple, Boston, Mass. Annual convention at Powers Hotel, Buffalo, N. Y., Sept. 23-26.

Lubrication — The Swain Company, 443 West 37th St., Chicago, Ill. Booklet, "Perfect Lubrication," describes the lubrication and lubricators manufactured by this company.

Turbines, Steam—Standard Turbine Corp., Scio, N. Y. Bulletin No. 8 covers the general points and description of turbines of single and multistage types in capacities up to 750 hp. manufactured by this company.

Steam Specialties—Locke Regulator Co., Salem, Mass. Catalog N describes the regulating devices for the conduct and control of steam and water pressure. Good illustrations and price lists of the different items add to the value of the catalog.

Motors, Polyphase—Century Electric Co., St. Louis, Mo. Bulletin No. 35 covers the squirrel-cage induction polyphase motors, one-sixth to 75 hp., Type "SC." Illustrations of the parts of the different-sized motors are a feature of the description.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal	Market Quoting	Aug. 11, 1924	Aug. 18 1924
Pool 1	New York	$2.30	$2.35
Smokeless	Columbus	2.10	2.15
Clearfield	Boston	1.90	2.35
Somerset	Boston	2.05	2.50
Kanawha	Columbus	1.40	1.55
Hocking	Columbus	1.55	1.65
Pittsburgh No. 8	Cleveland	1.85	1.90
Franklin, Ill.	Chicago	2.35	2.50
Central, Ill.	Chicago	2.10	2.25
Ind. 4th Vein	Chicago	2.35	2.50
West Ky.	Louisville	1.60	1.85
S. E. Ky.	Louisville	1.55	1.75
Big Seam	Birmingham	1.75	2.00

FUEL OIL

New York—Aug. 27, light oil, tank-car lots, 28@34 deg. Baumé, 4¾c. per gal., 36@40 deg. 5c. per gal., f.o.b. Bayonne, N. J.

St. Louis—Aug. 19, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.50 per bbl.; 26@28 deg., $1.55 per bbl.; 28@30 deg., $1.60 per bbl.; 30@32 deg., 4¼c. per gal; 32@36 deg., gas oil, 5c. per gal.; 38@40 deg., 6c. per gal.

Pittsburgh—Aug. 21, f.o.b. local refinery, 30@34 deg., fuel oil, 5¼c. per gal.; 36@40 deg., fuel oil, 5½c. per gal.; 34 deg., neutral, 8½c. per gal.

Dallas—Aug. 22, f.o.b. local refinery, 26@30 deg., $1.20 per bbl.

Philadelphia—Aug. 22, 28@30 deg., $2.05@$2.113 per bbl.; 18@22 deg., $2.075@$2.139; 13@16 deg., $1.78½@$1.848 per bbl.

Boston—Aug. 21, tank-car lots, f.o.b. heavy oil, 12@14 deg. Baumé, 4¾c. per gal., light oil, 28@32 deg. Baumé, 6¾c. per gal.

Cincinnati—Aug. 19, tank-car lots, f.o.b. local refinery, 24@26 deg., Baumé, 4¾c. per gal.; 26@30 deg., 4½c. per gal.; 30@32 deg., 5c. per gal.

Chicago—Aug. 18, 20@22 deg., 5½c. per gal.; 24@26 deg., 6c. per gal.; 28@30 deg., 6c. per gal.

New Plant Construction

Ark., Fort Smith—The City Water Commissioner plans to remodel plant and to purchase 250 hp. steam boiler.

Calif., Anaheim—The city voted $245,000 bonds for a municipal power plant.

Calif., Fullerton—The city is having cost estimates made for a new municipal power plant W. C. Record is city engineer.

Calif., Los Angeles—Beverly Hills Ice & Cold Storage Co., c/o H. C. Harns, Merritt Bldg., plans the construction of an ice and cold storage plant, estimated to cost, $100,000.

Calif., Los Angeles—Board of Supervisors of Los Angeles County, Hall of Records, will receive bids until Sept. 15th for furnishing all material, labor and equipment necessary for the erection and completion of boilers, framework and setting of boilers at the County Farm.

Ill., Chicago—Department of Public Works, Room 406 City Hall, is receiving bids until Sept. 19, for designing, fabricating, delivering and supervising erection of 2 complete steam-driven centrifugal pumping unis, one for Central Park Ave. pumping station, one for Springfield Ave. pumping station, estimated to cost $160,000. J. Ericson, 402 City Hall, is engineer.

Ill., South Chicago (Chicago P. O.)—The By-Products Coke Corp., 332 South Michigan Ave., Chicago, have retained Freyn, Brassert & Company, 310 South Michigan Ave., Chicago, as engineers for the design and construction of a new boiler plant of 2400 hp., to be installed at their coke plant here.

Ind., Elkhart—Board of Education plans the construction of a new power house at the Central High School, estimated to cost, $80,000.

Ky., Louisville—Louisville Hydro Electric Co. having preliminary surveys made by the H. M. Byllesby Co., 208 S. La Salle St., Chicago, for hydro-electric power plant.

La., Gentilly (New Orleans P. O.)—Louisville and Nashville R.R. Co., Louisville, Ky., plans yard improvements to include one 90-ft. electric-driven turntable, one electric-driven cinder conveyor, etc. Work will be done by owner's forces excepting the buildings which will probably be let by contract, bids to be taken about November. Estimated cost $600,000. W. H. Courtenay is chief engineer.

Mass., Boston—Gillette Safety Razor Co. awarded the contract for the construction of a 2-story, 70x75-ft. power plant on West First St., to Whidden & Beekman Constr. Co., 100 Boylston St., estimated to cost $45,000. C. T. Main Co., 200 Devonshire St., is engineer.

Minn., Owatonna—The Utilities Commission will receive bids until Sept. 16, for the construction of a complete underground steam-heating system in the business district. A. L. Mullergren, 555 Gates Building, Kansas City, Mo., is consulting engineer.

Mo., Aurora—Lawrence Water, Light and Cold Storage Co., is receiving bids for the construction of a 1-story, 67x90-ft. cold storage plant, estimated to cost $40,000. Heckenlively & Mark, 642 Landers Bldg., Springfield, are architects.

Mo., Bolivar—L. K. Green & Sons, Rich Hill, plans to increase the capacity of their power plant from 300 hp. to 1000 hp. and will also build a 22,000-volt, 3-phase transmission line with 30-ft. poles, etc.

N. Y., Batavia—Hoefler Ice Cream Co., 296 Connecticut St., Buffalo, is planning the construction of an ice cream plant on Swan St., estimated to cost $75,000. Refrigeration machinery will be required. F. A. Homelins, c/o Hoefler Ice Cream Co., is architect.

N. Y., Rochester—The Rochester Gas & Electric Co. is remodeling and extending the capacity of its power plant by the installation of three 875-hp. boilers.

Ohio, Cleveland—Cleveland Electric Illuminating Co. is taking bids for the construction of a 4-story, 115x238-ft. steam power plant at 1830 Lakeside Ave. Estimated cost $1,000,000. E. J. Cook, c/o owner, is engineer.

Ohio, Columbus—Rainbow Tire & Rubber Co., 187 South High St., C. E. Ross, president, will purchase three 25-hp. speed motors from 300 to 1200 rev.

Every one of these items is reported by our authorized correspondents who are instructed to verify every item sent in. This free weekly service is published in the interests of the buyer and the seller, to bring them together and get machinery moving.

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BUSINESS NEWS DEPARTMENT
Tenth Ave. at 36th St., New York

Okla., Cushing—City, L. A. Crewshaw, Mayor, will receive bids until Sept. 9th for the construction of earth dam and storage reservoir, 1,000,000 gal. water purification plant, pumping equipment, motor driven centrifugal pumps, mains, valves, hydrants, etc. Black & Veatch, Mutual Bldg., Kansas City, Mo., are engineers.

Pa., Elwood City—Elwood Water Co. awarded the contract for the construction of additions and alterations to pump station, boilerhouse and filtration buildings to J. W. McLaughlin, Park Bldg., Pittsburgh, estimated to cost about $25,000. J. W. Chester, Union Bank Bldg., Pittsburgh, is engineer.

Pa., Muncy—Muncy Water Supply Co., c/o F. M. Sprout, is receiving bids for a pumping station and a dam at intake, with a treating station.

Pa., Philadelphia—Philadelphia Grain Elevator Co., c/o Philadelphia & Reading Ry. Co. plans the construction of a grain elevator, including power plant equipment, three mechanical car unloaders, drying equipment, 4,000 to 5,000 hp. in motors etc. Estimated cost $2,500,000. Fegles Construction Co., 706 1st Ave., N. Minneapolis, is engineer.

R. I., Providence—Narragansett Electric Light Co. plans to install new boiler house and 30,000 kw. turbine and separate coal pulverizing plant. Two boilers have 20,252 sq.ft. of heating surface, and two have 18,581 sq.ft. of heating surface. Bids on the boiler house are expected August 30. Jenks & Bullon, 10 Weybosset St., are engineers.

Tex., Houston—The city voted $50,000 bonds for five new wells and pumps for water supply. J. C. McVea, 1318 Kipling St., is engineer.

Tex., Stephensville—Stephensville Ice & Cold Storage Co., will purchase a 15-ton ice machine and miscellaneous equipment.

Tex., Temple—Temple Ice Co., Box 58, will purchase a 5-ton ice machine and miscellaneous equipment.

Tex., Timpson—The City Commissioners will receive bids until Sept. 5th, for a sewage disposal plant, pumps, etc., estimated to cost $30,000. Elrod Engr. Co., 3124 Elm St., Dallas is engineer.

Tex., Vernon—The city, H. Mason, mayor, will receive bids until Sept. 9 for the purchase of one 300-hp. engine (oil burning) directly connected to 2300-volt, 3-phase, 60-cycle alternators with direct or belt-driven exciters, etc.

Tex., McAllen—Carroll College, c/o J. M. Carroll, 209 Stratford St., San Antonio, is having plans prepared for the construction of a 40 x 65 ft. power and heating plant in connection with the proposed new college buildings, estimated cost, $40,000. H. P. Smith, National Bank Commerce, San Antonio is engineer.

Vt., St. Albans—The Central Vermont Railway Co. awarded the contract for the construction of a mechanical coaling plant to R. Schaffer, Dearborn St., Chicago. Estimated to cost $55,000.

Wash., Olympia—G. A. Shamberger, Oroville, Okanogan Co., has filed application with M. Chase, Supervisor of Hydraulics for the purpose of developing 110,000-hp. plant. It is proposed to build dam 1150 ft. long at top, 1000 ft. at bottom and 46 ft. high.

Wis., Chili—Chili Co-operative Dairy Co., W. Lindow, Mgr., plans to purchase refrigeration machinery for the proposed new cheese factory, estimated to cost $40,000.

Wis., Madison—H. C. Buser, City Clerk is receiving bids until September 8th for 25,000 gal. steel water tank on 75 ft. tower, also 1,200 g.p.m. motor driven pump.

Wis., Milwaukee—O. Wollner, c/o Davis & Tuckwell, 377 National Ave., architects, plans to purchase, 80-hp. boiler and 40- to 50-hp. horizontal engine for proposed new laundry on Clybourne St., estimated to cost $45,000.

N. B., Fredericton—The City, A. Jennings, Chairman of Committee, Purch. Agt. plans to purchase one 300 hp. oil burning engine, two, 10 in. pumps, estimated cost. $30,000.

Ont., North Bay—The City Council proposes moving the pumping plant to another location, construct a new pump house and install additional electric pumps. Estimated to cost $100,000. W. Mackie, Town Hall, North Bay, is engineer.